Compressed Air Plant

THE PRODUCTION, TRANSMISSION AND USE OF COMPRESSED AIR, WITH SPECIAL REFERENCE TO MINE SERVICE

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PREFACE TO THE SECOND EDITION

THIS edition has been revised and substantially enlarged. Among the principal additions are some 90 pages of text and 63 illustrations, relating to the construction and operation of rock-drills, coal-cutting machines and channeling machines. This material is contained in Chapters XX, XXI, XXII, and XXIII. The detailed records of work of machine drills, in Chapters XX and XXI, I believe, will be found useful. Most of the data has not before been in print.

In Chapter III the theory of the compression of air is presented in greater detail, together with its applications to the operation and performance of compressors. The deductions of the more important formulæ are also given, such as those used for calculating the horse-power required for single- and multiple-stage compression. In this connection I desire to acknowledge the kind assistance of Professor Charles E. Lucke, of Columbia University, and Professor H. J. Thorkelson, of the University of Wisconsin. To Dr. Lucke my thanks are due for the use of his valuable, and hitherto unpublished, notes relating to the work cycles of air compression, with and without clearance. I would call attention also to the records of compressor tests in the latter part of Chapter X. These comprise a few typical tests, selected from a large number recently made by Mr. R. L. Webb, Mechanical Engineer, on compressors of different kinds in a well-known Canadian mining district.

Other new material has also been added, relative to the piston clearance of the air cylinders of compressors, and the ratio of inlet valve area to cylinder area. Numerous minor additions to the text have been made, together with corrections and alterations where required. The new matter aggregates some 135 pages of text and 87 illustrations. Many of the illustrations have been

furnished by the respective makers of the machinery, to whom credit is duly given. In preparing this revision I have kept in mind certain kindly criticisms and suggestions received from readers of the first edition. R. P.

NEW YORK, June, 1910.

PREFACE TO THE FIRST EDITION

THE increasing use of compressed air makes the subject of interest to practitioners in nearly all branches of engineering. Besides its more important power applications, such as the operation of rock-drills, air brakes, riveting machines, and railroad switching and signalling systems, the uses of compressed air are numerous in many minor branches of mechanical engineering, in caisson work and the construction of subaqueous foundations, and in manufacturing industries, chemical works, etc., where it serves a multitude of purposes entirely distinct from that of the transmission of power.

A realization of the breadth of the field has suggested that a book may be acceptable, addressed especially to those who are engaged in mining, tunnelling, quarrying, and other work involving the excavation of rock, with its concomitant operations. While the literature bearing upon this branch of compressed-air service is by no means small, it is for the most part scattered through the technical periodicals and transactions of engineering societies, and therefore not readily accessible to those who are out of convenient reach of engineering libraries. I am aware that little that is new can be said on this subject, and in writing the book I have availed myself freely of existing sources of information.

In the first part, I have endeavored to present a view of current practice as to the construction and operation of compressors.

Portions of the subject are dealt with at some length, for example: the types of compressor suitable for different kinds of

service, heat losses occurring in air compression, and the various forms of valves, valve-motions, and governing and unloading mechanisms, that constitute prominent features of modern compressor practice. A brief review is given of a few of the fundamental principles of air compression, but my intention has been to present only enough of the theory to make intelligible the formulæ employed for the ordinary calculations of the power and capacity of compressed-air plant, together with the questions concerning temperature changes, as affecting the production and use of compressed air. Many details of the design of compressors and proportions of their parts have been omitted, since these fall properly within the province of the mechanical engineer.

The second part is devoted to the applications (largely to mine service) of compressed-air transmission of power, including machine drills, pumps operated by compressed air, and mine haulage by compressed-air locomotives.

Many of the illustrations are reduced or adapted from working drawings kindly supplied by compressor builders. Others have been taken from catalogues of compressed-air machinery and from technical periodicals and books dealing with the different types. The origin of these has been stated in nearly every instance. My thanks are due to several of the technical journals, especially *Compressed Air Magazine* and *Mines and Minerals*, for many suggestions and in some cases for passages extracted either in substance or verbatim, from articles therein contained. For any important use or adaptation of published material, permission has been asked and obtained, and frequent references are given in foot-notes or in the body of the text. I also wish to acknowledge my indebtedness to Mr. Frank Richards's book on "Compressed Air," from which I have derived substantial assistance.

ROBERT PEELE.

SCHOOL OF MINES, COLUMBIA UNIVERSITY,
NEW YORK, May, 1908.

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PRODUCTION OF COMPRESSED AIR

CHAPTER I

INTRODUCTION

One of the most important applications of the transmission of power by compressed air is the driving of machine rock-drills; and to the necessity of providing for these drills a power medium suitable for use in mines and tunnels has been due, more than to any other cause, the development of the modern air compressor.

The time which has elapsed since the beginnings of this branch of engineering is short. The first percussion rock-drill, operating independently of gravity, was invented in 1849 by J. J. Couch, of Philadelphia. Though used only experimentally, it embodied the principal mechanical features of the modern machine drills, which have had such a striking influence in mining and tunnelling. Couch's machine, together with its immediate successors, such as the Fowle drill (1849–51) and the Cavé (Paris, 1851), were steam-driven and therefore unsuitable for underground work. In 1852, the physicist Colladon proposed the use of compressed air for operating rock-drills, in connection with the driving of the Mont Cenis tunnel, in the western Alps. His idea was developed by Sommeiller and others between 1852 and 1860, and in 1861–62 an air-compressor plant was first used successfully at that

tunnel. It was driven by water power and furnished air for ventilation as well as for the drills.

The transmission of power by compressed air thus dates from about the middle of the last century. It is hardly necessary to say that the early air compressors were crude in both design and construction. Sommeiller's first plant, though of large size and effectual in fulfilling its purpose, had some resemblance in principle to the old hydraulic ram, possessing no moving parts except the valves. Piston compressors, driven by steam engines, such as the Dubois-François, and more or less similar fundamentally to some of the wet compressors still in use, soon made their appearance. Probably the first compressors built in the United States were those employed at the Hoosac tunnel, in western Massachusetts, in 1865–66. The Burleigh, Norwalk, Clayton, and Rand compressors were among the earlier makes in this country.

But the Mont Cenis tunnel, about eight miles long and completed in 1871, the first connecting link through the Alps between the railway systems of France and Italy, was undoubtedly the field where were solved on a large scale the initial problems of compressed-air production and use; and to Sommeiller is due the honor of having laid the foundations of new practice, by which that great work was brought to a successful completion. From 1857 to 1861 the tunnel headings had been progressing slowly and in the face of great difficulties. Drilling was done by hand labor and blasting by black powder, the average advance for this period, in each of the two headings, being only about one and a half feet per day. At this rate, even granting that the work could have been finished at all by the means employed, over forty years would have been required to connect the headings and years more to complete the enlargement to full section. With machine drills, the speed of advance in each heading rose to four and three-quarters feet per twenty-four hours and later, when dynamite was introduced, to a little over six feet; this average being maintained for a period of six years.

Machine drills did not make their way into mining to any extent for some years after their successful application to tunnel driving.

It is difficult now to name the mining district in this country where they were first used, but their most important trial was probably in the Calumet and Hecla copper mine, Michigan. After strong and concerted opposition from the miners, the Rand drill was introduced there in 1878, and the value of machine drilling for hard ground was soon demonstrated by decreased costs of drifting and stoping and higher speeds of advance.

Compressed air has now a wide application in various branches of mechanical engineering and the arts and manufactures. In this book it is intended to deal only with its production and uses in connection with mining and tunnelling operations. Its two rivals in these fields of work are steam and electricity, regarding which a few general considerations may here be mentioned.

As compared with steam, compressed-air transmission of power is especially valuable and convenient for three reasons: *first,* its loss in transmission through pipes is relatively small; *second,* the troublesome question of the disposal of exhaust steam underground is avoided; *third,* the exhausted air is of some assistance in ventilating the working places of the mine. In large mines, where steam may be carried thousands of feet, down shafts and through lateral workings, for operating pumping engines, etc., the disadvantages attending its use become very apparent; the amount of condensation is serious, even when the piping is provided with good non-conducting covering, and the working efficiency falls to an abnormally small figure. Furthermore, aside from the heat produced by the use of steam, it is rarely feasible to employ efficient condensers for underground engines other than pumps, on account of the difficulty of obtaining the necessary condensing water and the additional space required. If the exhaust be discharged into the mine workings, even though they are large and well ventilated and the volume of the exhaust steam comparatively small, the temperature and quantity of moisture in the air would be considerably increased. Deterioration of the timbering is thereby hastened, the roof and walls of the workings are softened and slacked off, especially in collieries, and the mine atmosphere is rendered oppressive and unwholesome. The presence of hot steam

pipes in confined workings, or in the narrow compartments of shafts, is also objectionable.

Although the loss from condensation in long steam lines may be diminished by covering the pipe with efficient non-conducting material, still, even with the best covering, the effective pressure at a distant underground engine is greatly reduced, and very uneconomical working is the result. On conveying steam a distance of several thousand feet the pressure may be reduced to half the boiler pressure, or even less. For example, in the case of a pump, or other engine, situated 2,000 feet from the boiler and using 200 cubic feet of steam per minute at a boiler pressure of 75 pounds, with a four-inch mineral-wool-covered pipe, the effective pressure at the engine would be only about 58 pounds; or, with a poor covering, like some of the asbestos lagging often used, it might easily be as low as 35 pounds. For compressed-air transmission, on the other hand, the reduction of pressure for the same volume of air, size of pipe, and initial pressure, would be 9.3 pounds, giving a terminal pressure of 65.7 pounds. However, as the speed of flow in pipes for economical transmission is greater for steam than for air, a comparison based solely on piping of the same diameter cannot justly be made. In the above example, if the diameter of the pipe were smaller the gain in reduced radiation would outweigh the increased frictional loss, and the net loss would be diminished. Since the frictional loss varies inversely, and the loss from radiation directly, with the diameter, the size of the steam pipe can be so proportioned as to produce a minimum loss under the given conditions. With compressed air the case is different, since the question of radiation is eliminated. If the diameter of the pipe be increased to 5 inches the loss of pressure, or head required to overcome friction, is reduced to 2.8 pounds and increasing the distance to one mile it would be only 7.4 pounds. Furthermore, the increased cost of the larger air pipe would be offset by the expense of the non-conducting covering necessary for steam transmission.

Thus, compressed air may be conveyed long distances with but small loss of pressure, and is readily distributed for application to a variety of underground uses, for which steam is not practicable.

Compressed air is always ready to do its work, and, aside from leakage of transmission pipes, which is in large measure preventable, suffers no loss nor diminution of power when not in actual use. For performing work *intermittently*, at a distance from its source, it is therefore particularly valuable, because the air pressure is maintained nearly constant during intervals of work, without further expenditure of power. With steam transmission, on the contrary, power is continually dissipated by radiation, whether in use or not, and a steam engine, when stopped for any length of time, loses much of its normal working temperature and becomes a receptacle for water of condensation.

Though in mining compressed air is employed mainly for operating machine drills, other applications are found in the driving of underground hoists and pumps in confined workings. Mechanical coal cutters, for mining bituminous coal, are sometimes operated by compressed air, and the employment of compressed-air locomotives in mines and extensive tunnelling operations furnishes an example of its capacity for storing power, in contradistinction to its function as a power transmitter. The introduction of compressed-air drills has facilitated the rapid driving of long railroad and mining tunnels, which otherwise would have been greatly delayed or completed only with extreme difficulty. Had compressed-air power, together with the high explosives, not been available, it may well be doubted whether the great tunnels through the Alps and elsewhere, and the numerous long mine tunnels driven in recent years in this country, would have been at all practicable.

Without attempting to review in detail the comparative merits of electricity and compressed air, it may be pointed out that the application of electricity for transmitting power in mines has increased enormously in importance during the past twenty-five years. The peculiar requirements of mine service have been in nearly all cases successfully met by modifications and adaptations of standard forms of electric apparatus. Both means of power transmission possess characteristics which adapt them particularly for underground work. But, although by virtue of its numerous successful applications, electricity has become a rival of compressed air in most

branches of mine work, their spheres of usefulness are not identical and the field is broad enough for both. It is often stated that the first cost of an electric plant is lower than that of an equivalent compressed-air plant. A broad generalization, however, does not cover the case. There is actually but little difference between the costs of the power plants themselves, the advantage being generally with the compressor. Considering the question of the transmission of a given power, the cost of the electric conductor line for short distances is much less than that of compressed-air pipe; but the cost of the electric line increases as the square of the distance, while the cost of the pipe line increases only as the first power of the distance. Hence, a point is soon reached where compressed-air transmission becomes the cheaper. It is in the greater efficiency of generation that electric power has the advantage.

In one direction only has electricity failed hitherto to meet every requirement. While compressed-air drills, though far from being economical considered simply as machines, nevertheless admirably fulfil their purpose, no perfectly satisfactory electric rock-drill has yet been produced. However, this problem has for years been receiving much attention from electricians, both in this country and abroad, and there is reason to anticipate its successful solution in the near future. The Temple "electric-air" drill, brought out some four years ago, and already well tested under a variety of conditions, may be referred to here as a remarkably efficient and ingenious machine, but it is not an electric drill in the ordinary meaning of the term. It is rather a combination of a compressed-air drill, operated by a small, electric-driven compressor which is mounted on a truck close to the drill itself. As there is no exhaust, the same air being used over and over, one of the incidental advantages of the ordinary air drill is missing, namely, that of assisting somewhat in ventilating the mine workings, in those places where ventilation is most needed. Keeping this in mind, together with such minor uses of compressed air as the cleaning of drill holes preparatory to charging, and driving out the smoke of blasting from working places, it seems doubtful whether, for underground mining, electric drills of any kind can be expected to supersede entirely those oper-

ated by compressed air. Given the necessity for a compressed-air plant for the rock-drills, as is the case in most metal mines, it may often be more advantageous to provide the additional compressor capacity required for driving underground pumps, hoists, and other machines as well, than to erect a separate and distinct plant for generating electricity.

Because of the view usually taken of the lack of economy in the operation of compressed-air drills, it has been customary in the past to consider compressed air in general as a form of power respecting which the questions of convenience and expediency are more weighty than the attainment of a high degree of efficiency. In recent years, however, as the principles of air compression have become better understood, a substantial improvement has taken place, not only in the design of the compressors themselves, but also in the installation of pipe lines and in the operation of the machines using the compressed air. The consequences of overloading a compressor, and thereby driving it beyond its proper speed, are now comprehended by every intelligent master mechanic as being wholly different from those produced by overloading a steam engine. The results of leaks in air pipes, and of using air mains of too small a diameter, are also understood and avoided. Better practice prevails in the field, and in the production, transmission, and use of compressed air a much higher total efficiency is now realized than was formerly thought possible.

CHAPTER II

STRUCTURE AND OPERATION OF COMPRESSORS

An air compressor consists essentially of a cylinder in which atmospheric air is compressed by a piston, the power for driving which may be derived from a steam engine, water-wheel, or electric motor. The air cylinder is almost invariably double-acting, and as such is provided with inlet and discharge or delivery valves in each cylinder head. On the forward stroke the air is compressed by the advancing piston, while the decrease in pressure, or, as it is commonly termed, the tendency to form a vacuum, behind the piston causes the inlet valves to open under atmospheric pressure, thus allowing the outside air to flow into the cylinder. At each stroke a certain volume of compressed air is forced from the cylinder through the discharge valves, into a pipe leading to a large reservoir or receiver, whence the air enters the transmission pipe or main.

Before considering the operation and various appurtenances of the air and steam cylinders, it will be well to examine the general mechanical structure of the compressor and the modes of applying the power. Probably no single classification of air compressors can be made sufficiently comprehensive to present intelligibly all of their salient features. In attempting a classification three widely different bases of comparison suggest themselves. *First*, several clear distinctions result from a consideration of the general structural characteristics of air compressors regarded purely as engines; *second*, they may be classed according to the mode of dealing with the heat necessarily produced during compression of the air; and *third*, the numerous and varying types of valves and valve-motions devised in modern practice for controlling the distribution of the air in the compressing cylinders constitute a basis for comparison

which, though not so simple as the others, is in some respects quite as useful and important.

The first classification only will be given here, the others being taken up respectively in Chapters IV to VI and VII to IX. Air-brake and gas compressors, vacuum pumps and other special forms of air-compressing machinery are not included, as this book will not deal with compressors other than those which are applicable to mine service.

Under the first classification and taking the steam-driven compressor as the type form, four subdivisions may be named:

1. **"Straight-line" Compressors.** In these, which are made by all builders, the steam and air cylinders are set tandem on a common piston-rod. They are always provided with a pair of fly-wheels, one on each end of the crank-shaft, which are driven by outside connecting-rods from a cross-head between the steam and air cylinders. Their structural form is thus simple and compact to a marked degree. Figs. 1 and 2 illustrate a Laidlaw-Dunn-Gordon straight-line compressor, with Meyer valve gear, a section of the steam cylinder being shown in the plan and of the air cylinder in the elevation. Fig. 3 is a perspective view of an Ingersoll-Rand straight-line compressor.

2. **Duplex Compressors.** (*a*) Two engines are placed side by side, each being complete in itself and consisting of tandem steam and air cylinders, with their cranks set at 90 degrees on a common fly-wheel shaft. Each side of the duplex is in effect a straight-line compressor. They are almost invariably horizontal, and the steam cylinders are always nearest the crank-shaft. Figs. 4 and 5 show a type of the duplex compressor. (*b*) One air and one steam cylinder, side by side, with a common crank-shaft, may in one sense be classed as duplex, though its operation is entirely different from (*a*) in the disadvantageous distribution of the load. While obsolete in America, this form is occasionally adopted by some European builders for special purposes. It is not well balanced, occupies at least three times the floor space of a straight-line compressor of the same capacity, and requires more expensive foundations.

3. **Compressors with Compound Steam Ends.** (*a*) Duplex, horizontal, cross-compound; a simple or single-stage air cylinder being set tandem to each steam cylinder. This form is now rarely used. The considerations leading to the compounding of the steam end make it desirable to adopt stage compression for the air end. (*b*) Vertical compound; the air cylinders being placed respectively above the high- and low-pressure steam cylinders. This also is a somewhat unusual design. It is advantageous in saving floor space, though this consideration is rarely of consequence at mines. As an example, the King-Riedler compressor may be cited (Fig. 6).* Some very large plants of this type, up to a capacity of 8,000 cu. ft. of free air per minute, have been built for South African mines. Vertical compressors have the disadvantage of being complicated and difficult to maintain and repair.

* From *American Machinist*, Oct. 16th, 1902, p. 1,475.

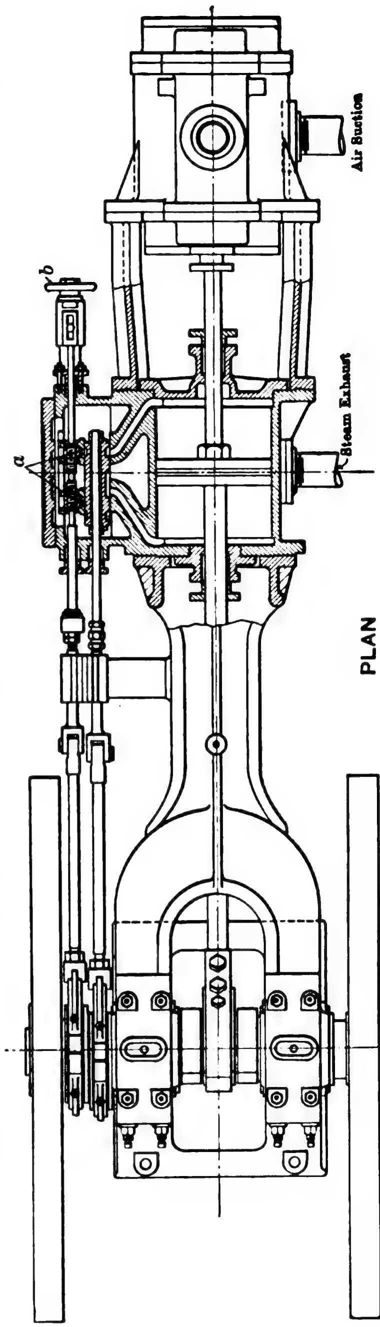


FIG. 1.—Laidlaw-Dunn-Gordon Straight-Line Compressor.

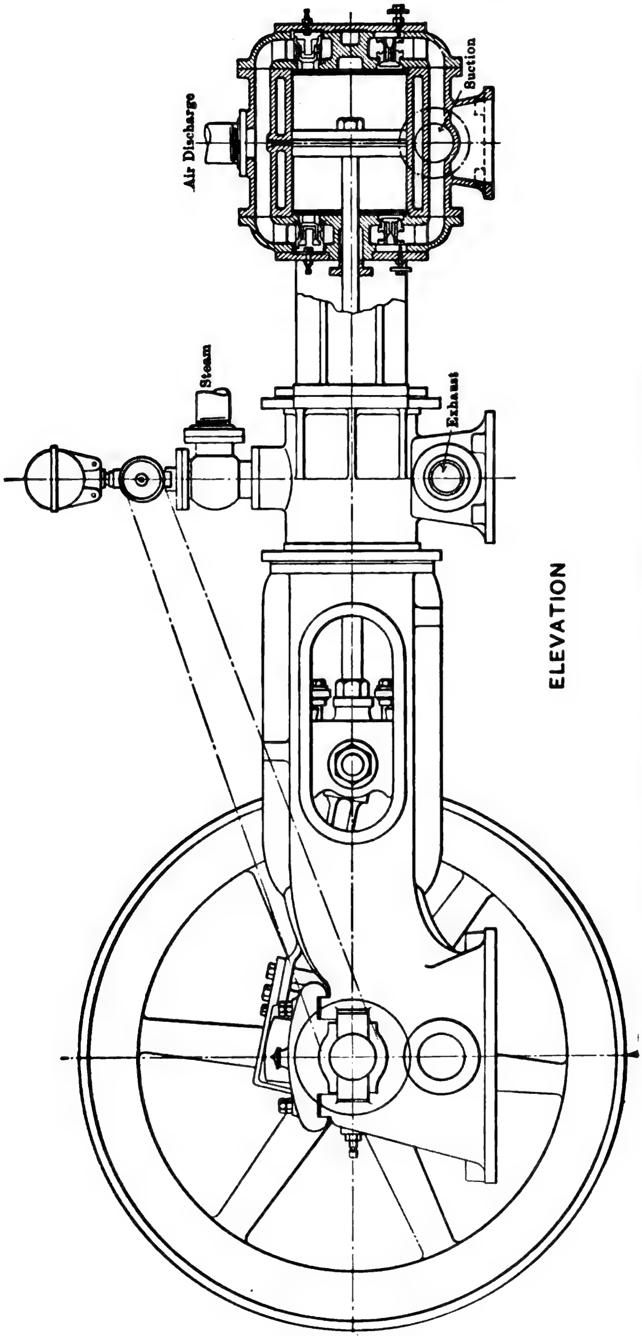


Fig. 2.—Laidlaw-Dunn-Gordon Straight-Line Compressor.

4. **Compound or Stage Compressors,** in which the air cylinders themselves are compounded. The air end may be of the double-, triple-, or quadruple-stage type, according to the air pressure to be produced.* Stage compressors are now made by nearly all builders in the United States, and compose the most important class of air-compressors for general use. (*a*) Straight-line form, as in (1). These have two-stage air ends, some having compound steam ends also. Fig. 8 shows the longitudinal section of a Norwalk compressor with compound steam cylinders, and Figs. 7, 9, 10, 11 and 12, respectively Ingersoll-Rand, Norwalk, Leyner, and Sullivan compressors, with simple steam cylinders. (*b*) Duplex steam end, with two-stage air cylinders. A longitudinal section of a Sullivan compressor of this class is given in Fig. 14, and a perspective view of a recent type of Leyner compressor in Fig. 16. (*c*) Duplex, cross-compound steam end, with two- to four-stage tandem air cylinders. These are designed for large plants only. The two-stage type is widely used. Those having air cylinders of more than two stages are for special high-pressure service, such as furnishing air for underground compressed-air locomotives. Figs. 15 and 21 are perspective views of two of the latest designs of the Ingersoll-Rand cross-compound, two-stage compressors, class O. Figs. 17 and 18 show the general plan and side elevation of a Riedler, and Figs. 19 and 20 similar views of an Allis-Chalmers Corliss compressor of this class; Fig. 22 is a perspective view, and Fig. 23 a reproduction of a working drawing, in plan and elevations, of a Laidlaw-Dunn-Gordon compressor, which will further illustrate this type.

As based on structural characteristics, compressors may also be classified as: (*a*) *Direct-driven* by steam- or water-power—the motor end being directly connected with the air cylinders. Among water motors the bucket or impulse wheels are best adapted to this service; (*b*) *Belt-driven* from independent motors: steam-engines, water-wheels, or electric motors. These are built by most of the American makers, and are in common use for mine and other

* It may be noted that the Norwalk Iron Works Co. was the pioneer in the field of stage compression, having begun in 1880–81 to build this type of compressor for ordinary service.

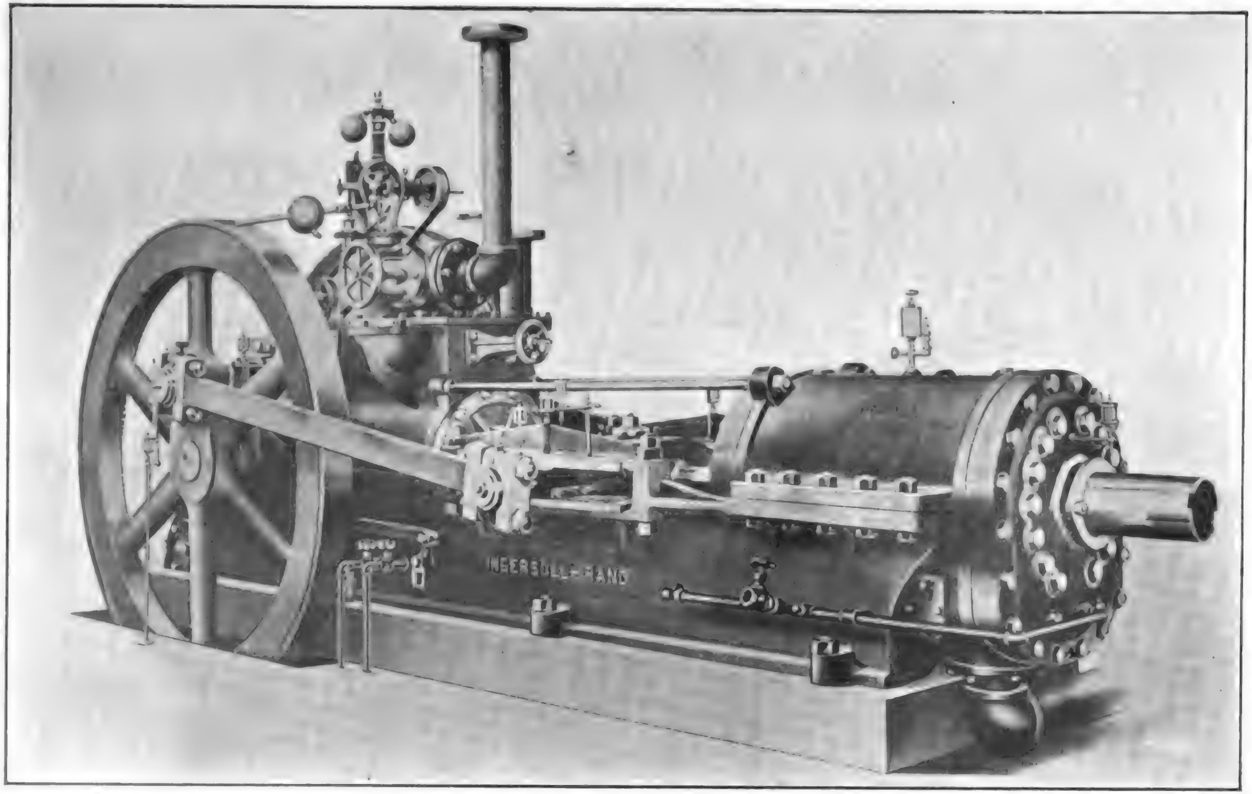


FIG. 3.—Ingersoll-Rand Straight-Line Compressor, Class A—1.

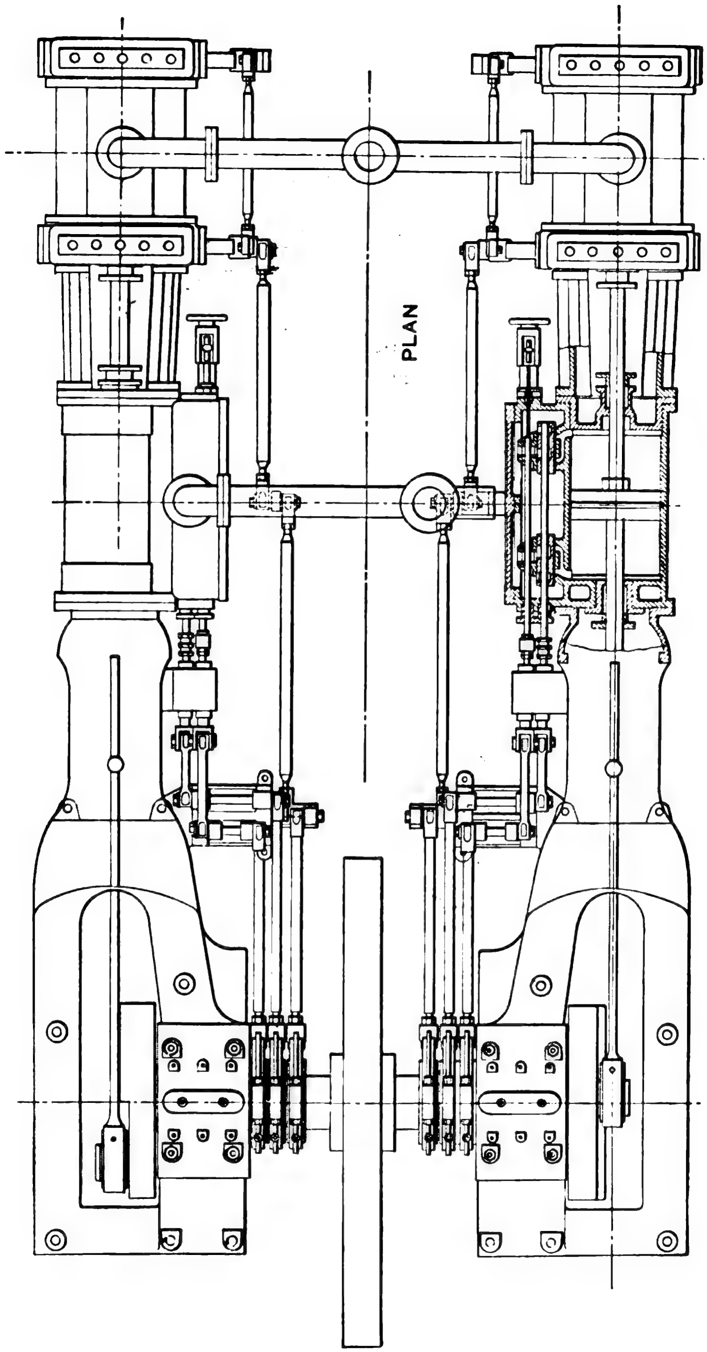


FIG. 4.—Laidlaw-Dunn-Gordon Duplex Compressor.

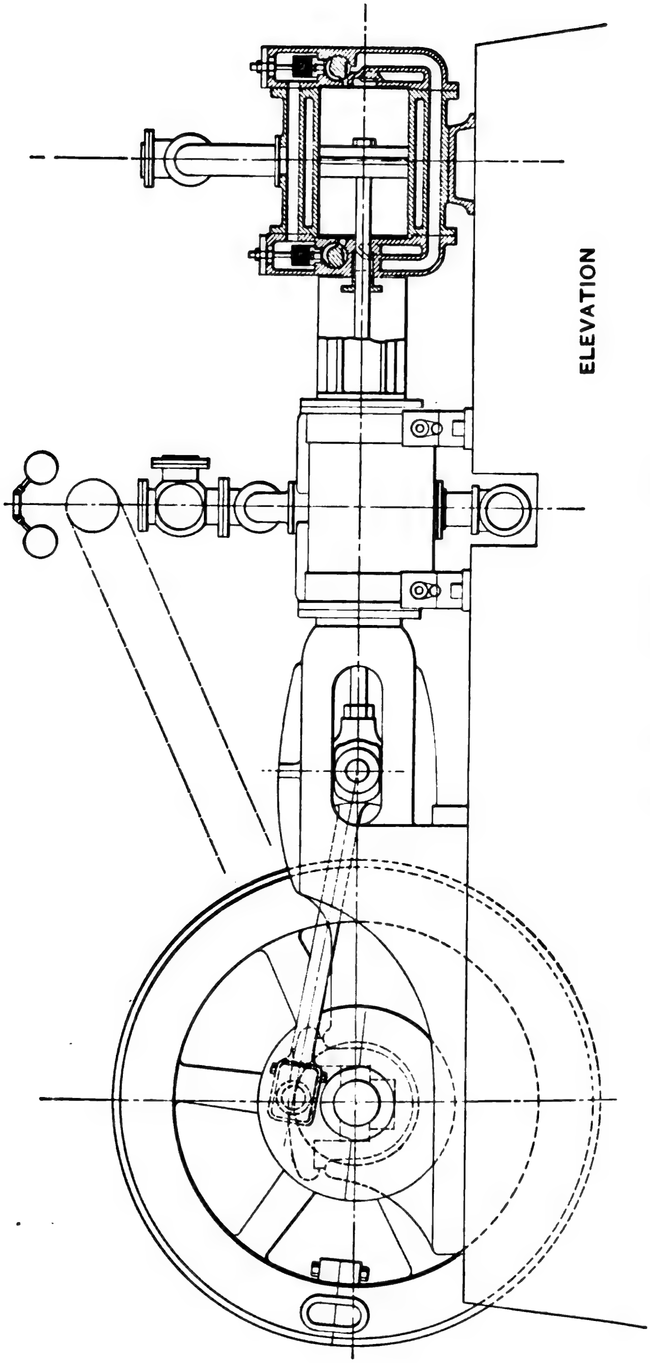


FIG. 5.—Laidlaw-Dunn-Gordon Duplex Compressor.

service. Chain-driven and direct-geared compressors are also occasionally employed, as noted hereafter.

So-called "half-duplex" compressors are furnished when re-

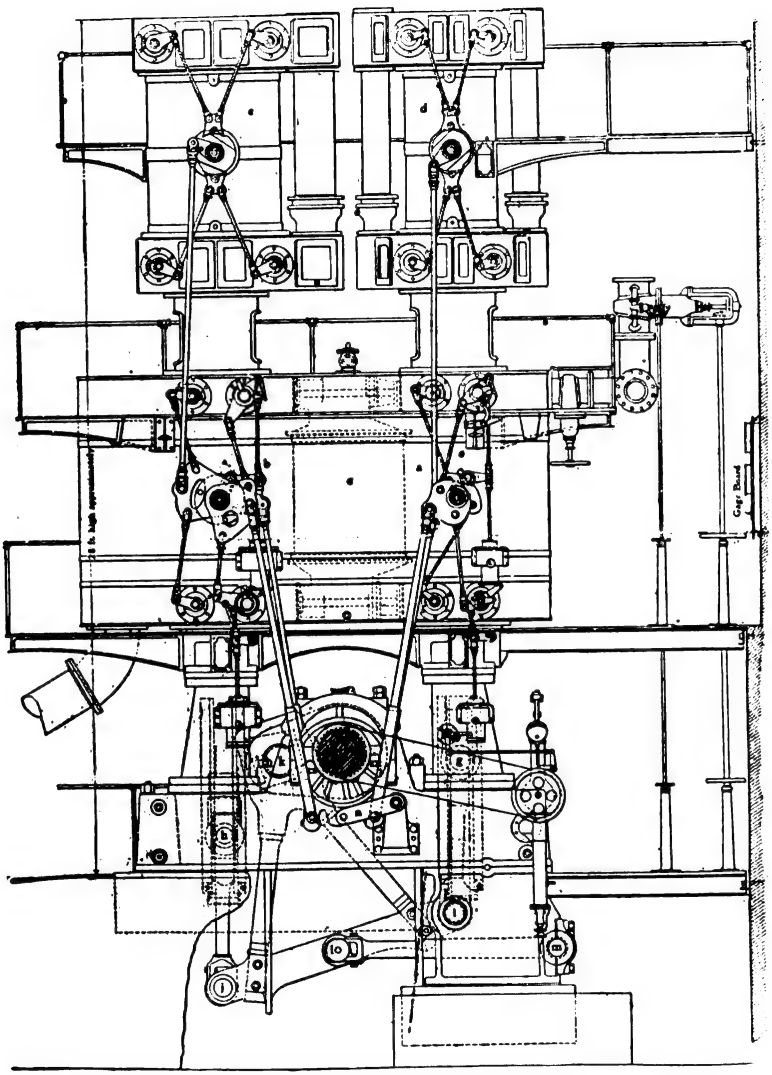


FIG. 6.—King-Riedler Compound Vertical Two-Stage Compressor.

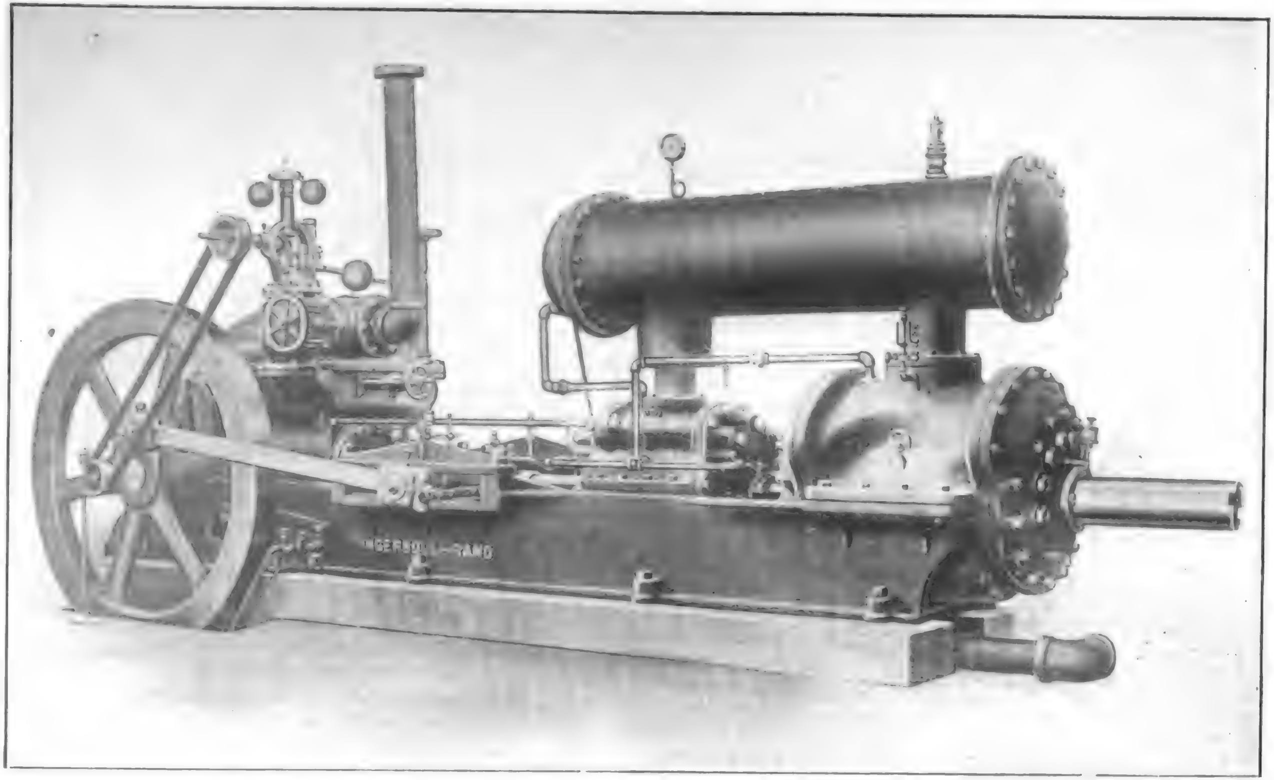


Fig. 7.—Ingersoll-Rand Straight-Line, Two-Stage Compressor, with Simple Steam End.

quired. They consist of either the right- or left-hand half of a duplex compressor, an extended crank-shaft and out-board pillow-block being provided temporarily. An advantage of this form is that, if only a comparatively small quantity of air is needed for a time—as during the development of a mine or the sinking of a shaft—one-half of a duplex compressor may be installed at first, the second half being readily added when required. The capacity is thus doubled at a moderate cost.

Comparison of Types of Compressor

The **straight-line compressor** is largely employed for rather small plants or for temporary service. It is compact, strong, and self-contained, the entire engine being carried by a single bed-frame and requiring a relatively inexpensive foundation. The floor space occupied is much less than for the duplex form. The air and steam cylinders are just far enough apart to allow the cross-head and guides to be placed between them. From the cross-head the fly-wheels are driven by connecting-rods on each side. By using a pair of fly-wheels each is made smaller and lighter than if there were but one, and the moving parts are better balanced. While useful for moderate air pressures and fairly constant loads, and satisfactorily filling an important field of work, the straight-line compressor is not capable of operating with the steam economy desirable and even essential in plants of large capacity; nor is it self-regulating at much less than, say, forty per cent. of its full load. These compressors are usually made of capacities from the smallest up to 1,700 or 1,800 cu. ft. of free air per minute, the last-named sizes developing from 275 to 300 horse-power. Further details of the operation and distribution of load in these compressors are given on page 32.

The **duplex compressor** is always preferable to the straight-line for large plants. It is better adapted to varying loads, arising from differences of air pressure, because the resistance is more uniformly distributed throughout the stroke. By reason of its quartering cranks it may be run at extremely slow speeds without stopping on a center; and it is self-regulating and capable of deal-

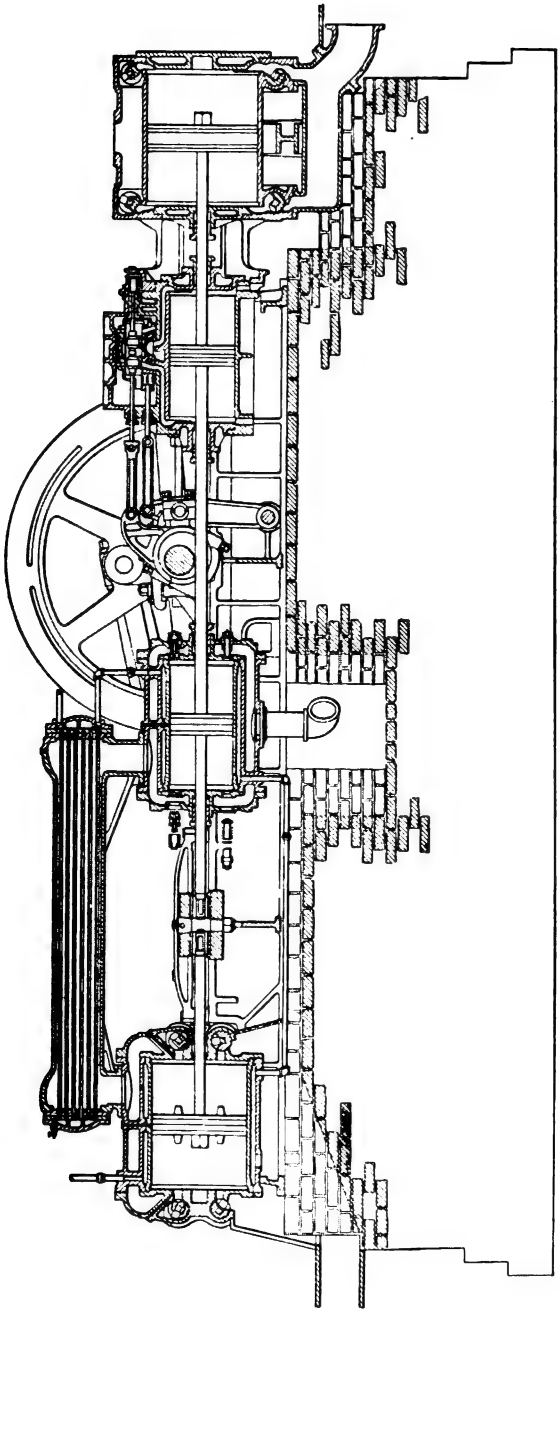


FIG. 8. —Norwalk Compound Straight-Line, Two-Stage Compressor. Longitudinal Section.

ing economically with a range of load down to considerably less than one-quarter or one-third of its normal. As a rule, the friction loss (total horse-power consumed by friction of the engine) of the duplex compressor is no greater and is often less than that of a straight-line of the same capacity. For large Corliss compressors, in good order, this loss may be put at not over five to seven per cent.* While these figures are sometimes equalled by the best

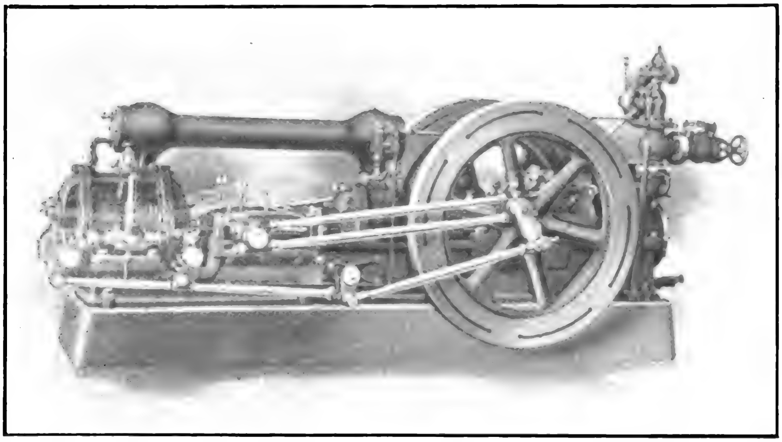


FIG. 9.—Norwalk Straight-Line, Two-Stage Compressor, with Simple Steam End.

straight-line compressors, it is safe to say that the loss in the latter is generally higher.

Of late years the Corliss type of engine has come into general use for driving large duplex compressors, especially when compounded in both steam and air end, as its valve gear is well adapted for dealing with the variations of air pressure under which compressors are usually called on to work. By the majority of builders the Corliss valve gear is employed, at least for large plants, for the air as well as the steam cylinders.

The foundation of the duplex compressor is necessarily more

* In this connection, see an article by J. Parke Channing, in *Mines and Minerals*, May, 1905, p. 475, containing the results of an efficiency test on a 300-H.-P. compound, two-stage Nordberg Corliss compressor, at the Burra-Burra mine of the Tennessee Copper Co. Its efficiency was found to be 78.1 per cent. total. The horse-power consumed by friction was only 5.2 per cent.

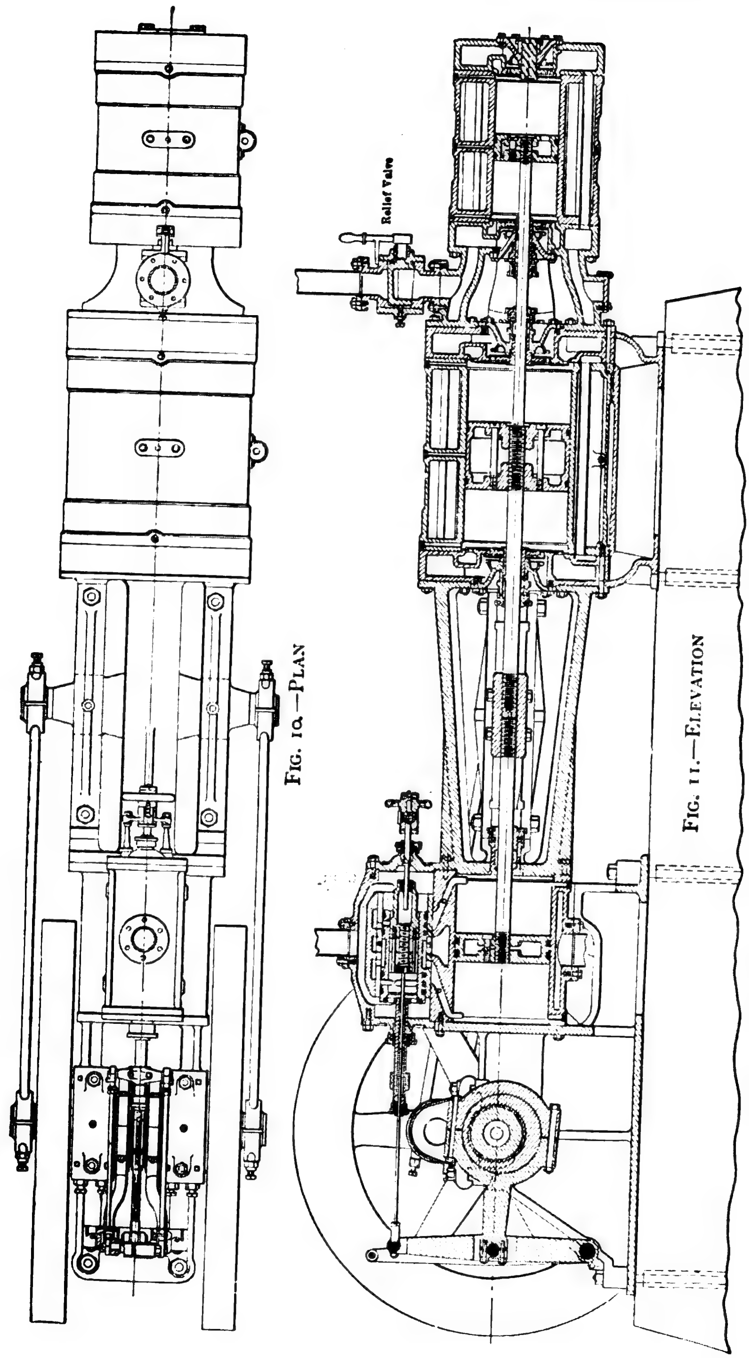


FIG. 10.—PLAN

FIG. 11.—ELEVATION

FIGS. 10 and 11.—Leyner Straight-Line, Two-Stage Compressor.

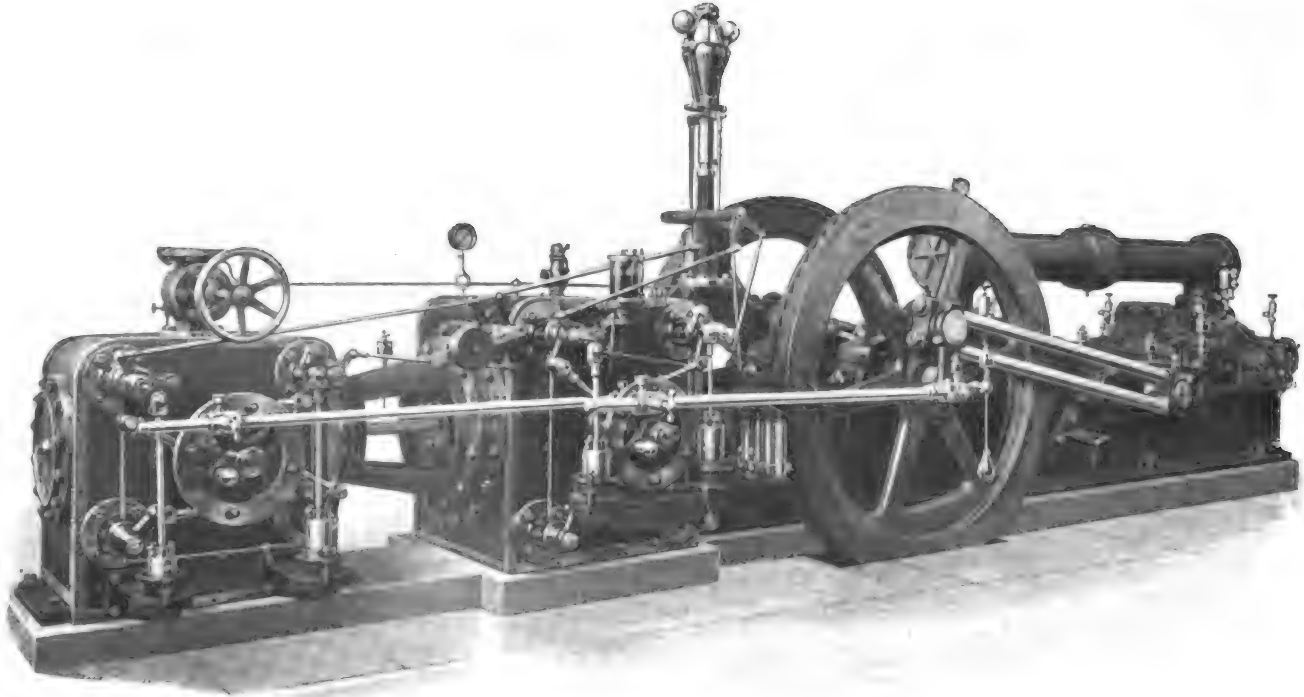


FIG. 13.—Sullivan Corliss, Tandem Compound, Two-Stage, Straight-Line Compressor, Class "W C."

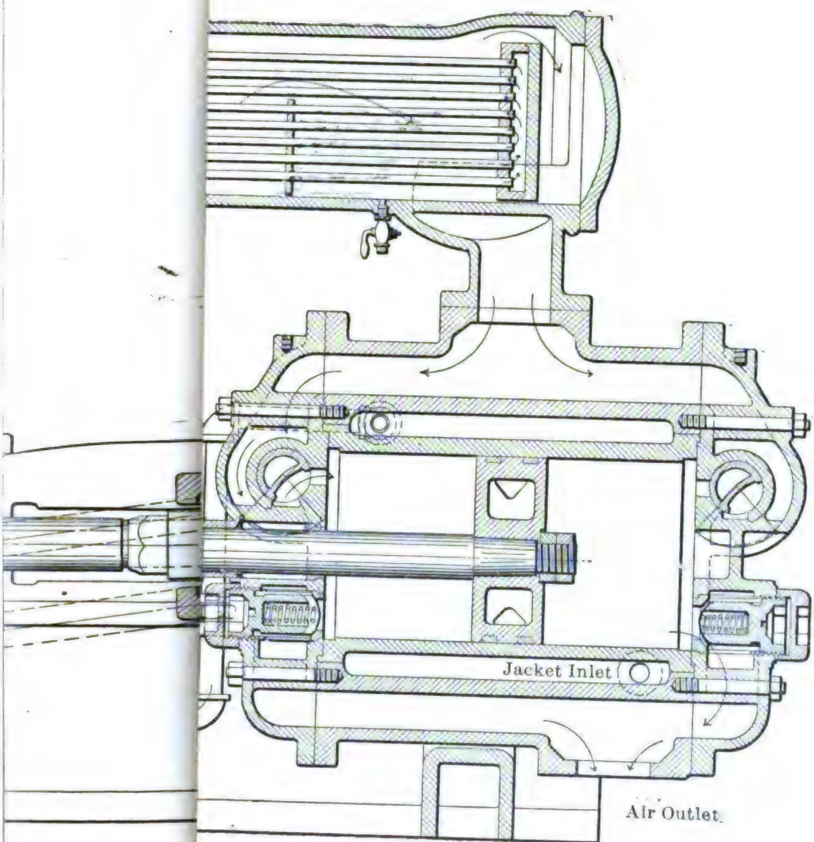


FIG. 12.—

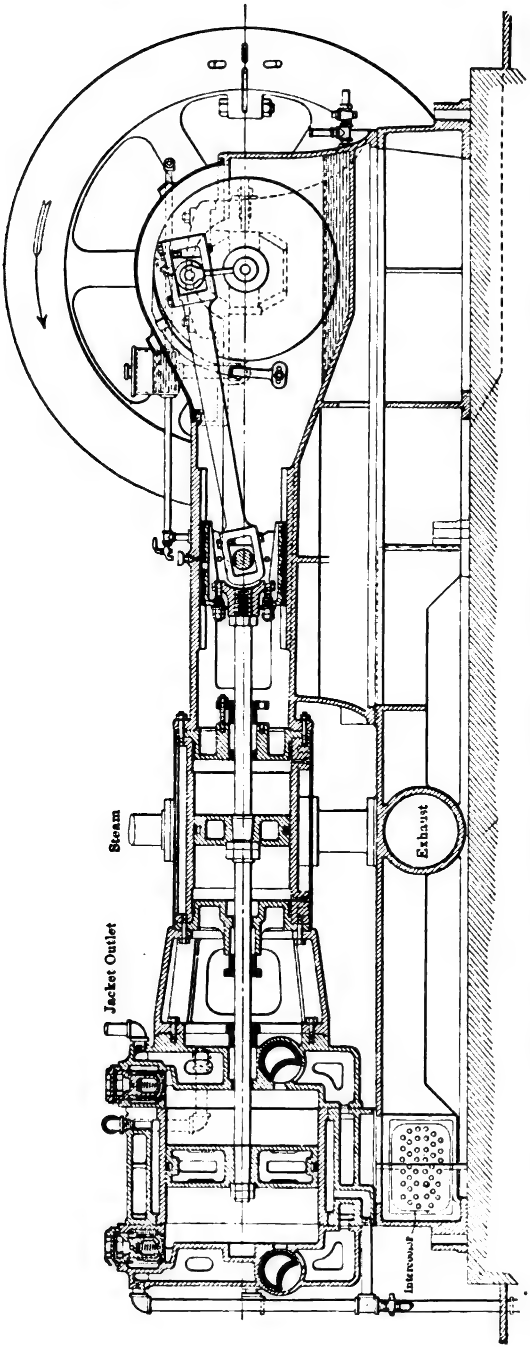


FIG. 14.—Sullivan Duplex, Two-Stage Compressor. Longitudinal section through low-pressure cylinder.

expensive than that of the straight-line, and must be substantially built if perfect alignment is to be maintained. Each pair of cylinders are solidly connected, either by trunk-frames or heavy tie-bolts. A complete girder-frame may be provided (Figs. 10, 14) to avoid any possibility of movement. The tandem steam and air cylinders on each side are best placed far enough apart to prevent the same portion of the piston-rod from passing alternately into each stuffing-box. The reasons for this are: *first*, the piston-rod is apt to wear differently in the two stuffing-boxes, so that it becomes difficult to keep them well packed and tight; *second*, in this construction the steam and air piston-rods are made in separate parts, coupled together between the cylinders. This is a matter of convenience in making repairs, when it becomes necessary to take the compressor to pieces; also, the air valves, when of the poppet form and in the cylinder head, are more accessible. An incidental advantage of the duplex compressor is that, as each half is complete in itself, one side may be disconnected for repairs or when a smaller capacity is temporarily desired.

Compressors with Compound Steam Cylinders. The advantages in point of economy secured by compounding the steam end of air compressors are even more striking than in the case of ordinary stationary engines, for two reasons: *First*, because the conversion of power from one form to another is necessarily attended by some loss, and should therefore be conducted as economically as possible; *second*, because, as will be shown hereafter, the operation of compressing air involves particularly unfavorable load conditions. The valuable features of the duplex compressor become most apparent when the steam cylinders are compounded and furnished with a proper condenser. In plants of any size, a steam saving of, say, twenty per cent. may thus be readily attained, not only by getting the full expansive power out of the steam, but also by avoiding frequent loss of power due to imperfect speed regulation and consequent blowing off of air at the relief or safety valve.

Stage Compressors in recent years have come into general use for mining and other service. It is now recognized that even for ordinary pressures of, say, seventy-five pounds, such as are com-

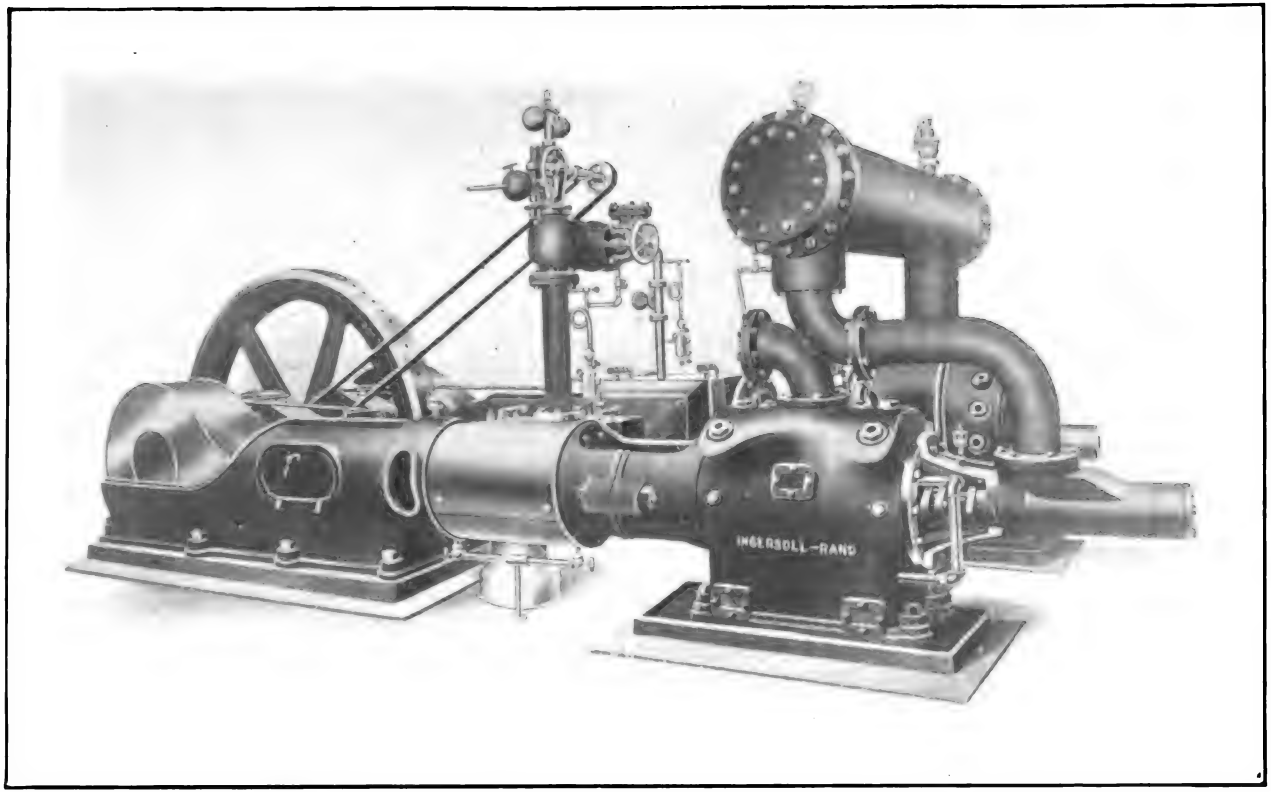


FIG. 15.– Ingersoll-Rand Cross-Compound, Two-Stage Compressor, Class "O."

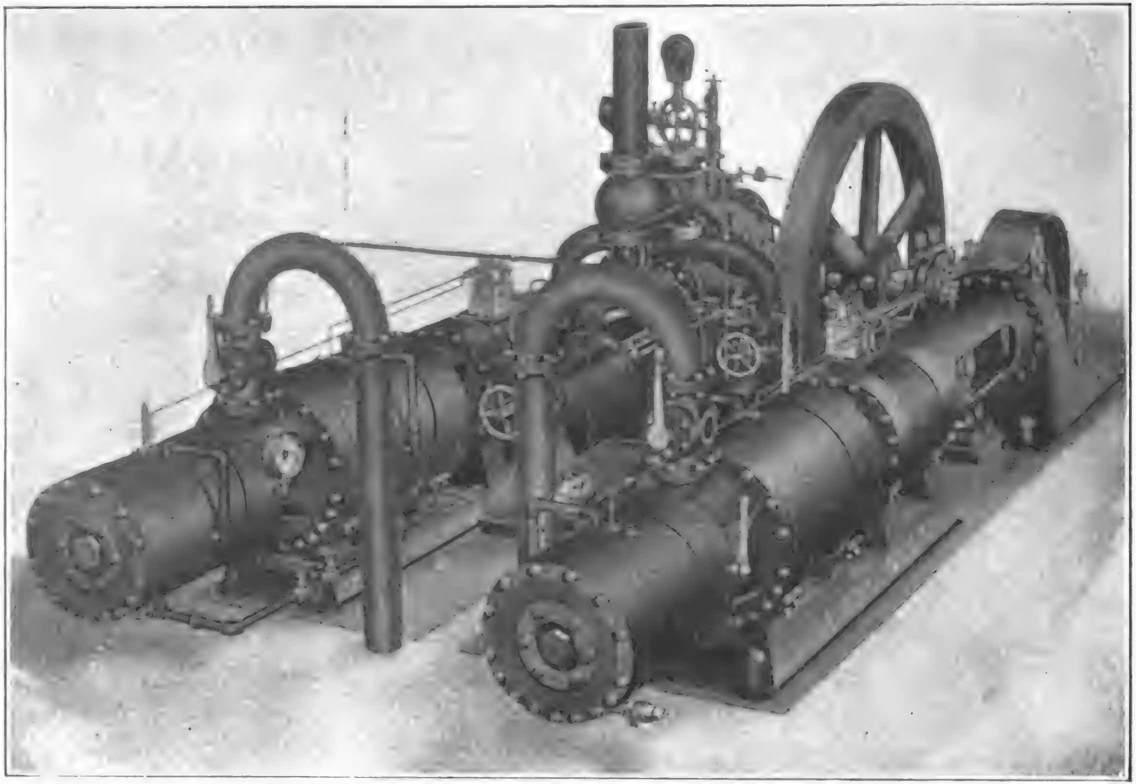


Fig. 16.—Leyner Duplex, Two-Stage Compressor, with Simple Steam Cylinders.

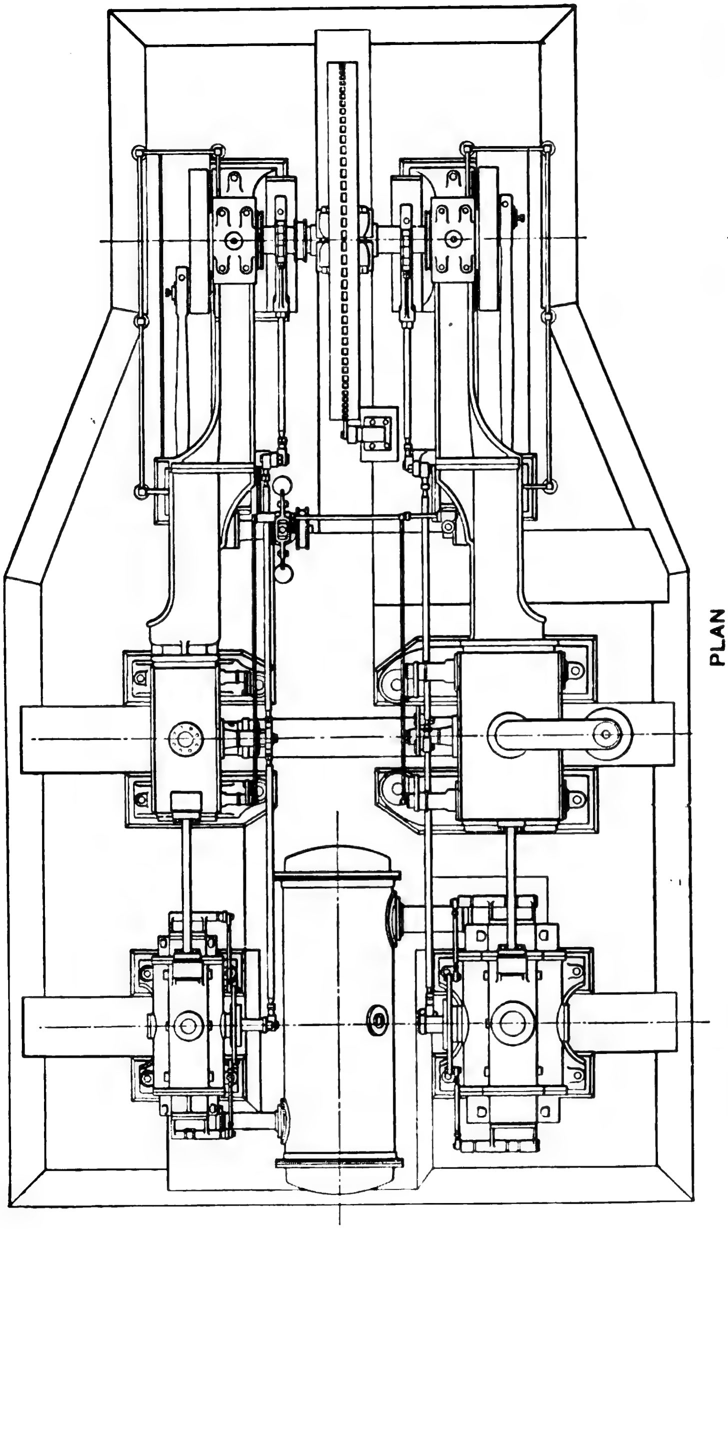


FIG. 17.—Riedler Cross-Compound, Two-Stage Compressor. 14" and 24" × 36" steam and 15" and 24" × 36" air cylinders.

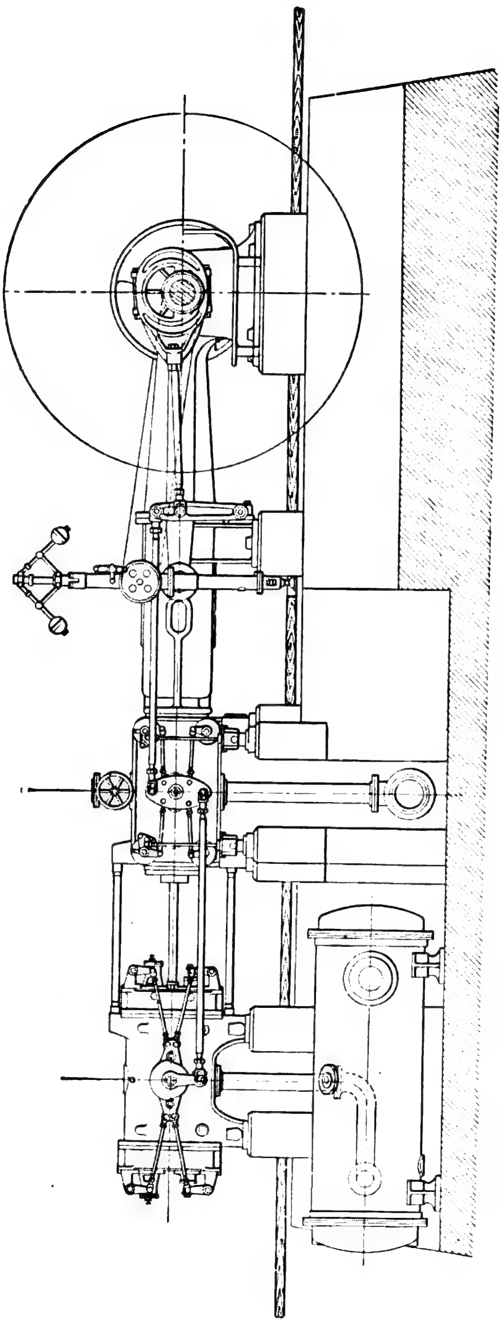
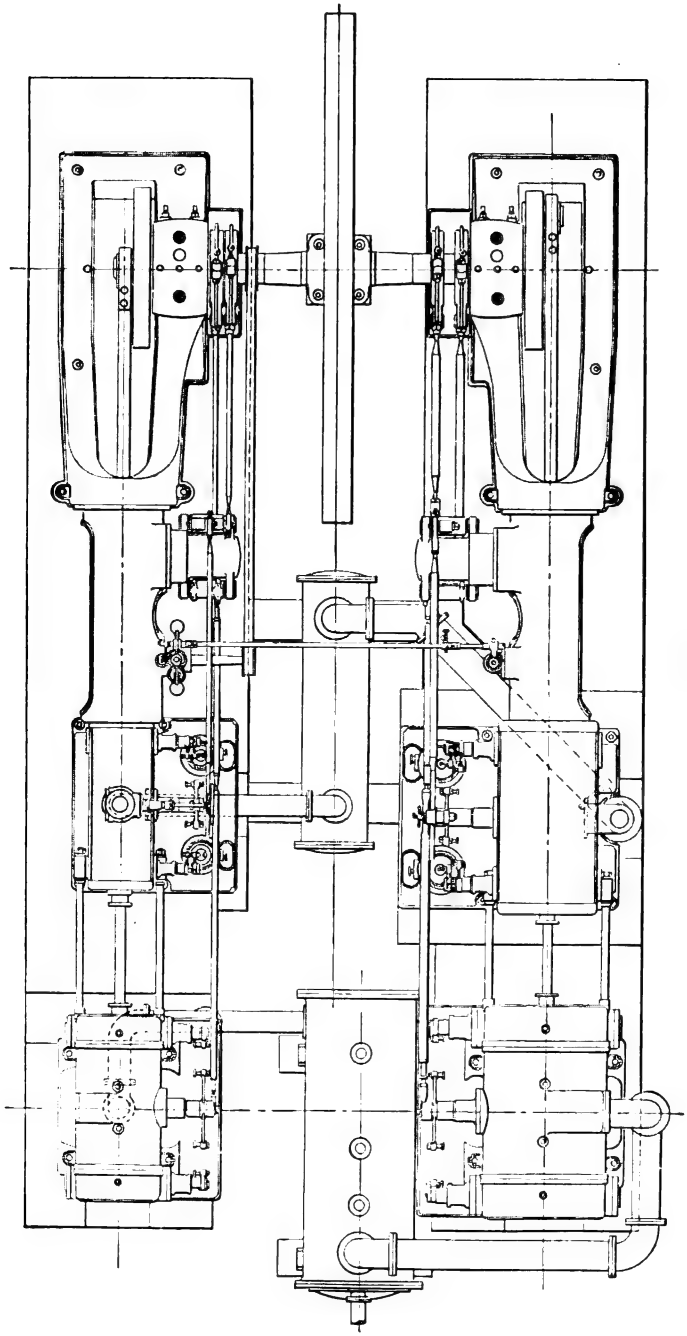


FIG. 18.—Riedler Cross-Compound, Two-Stage Compressor.



PLAN

FIG. 19.—Allis-Chalmers Cross-Compound Corliss, Two-Stage Compressor.

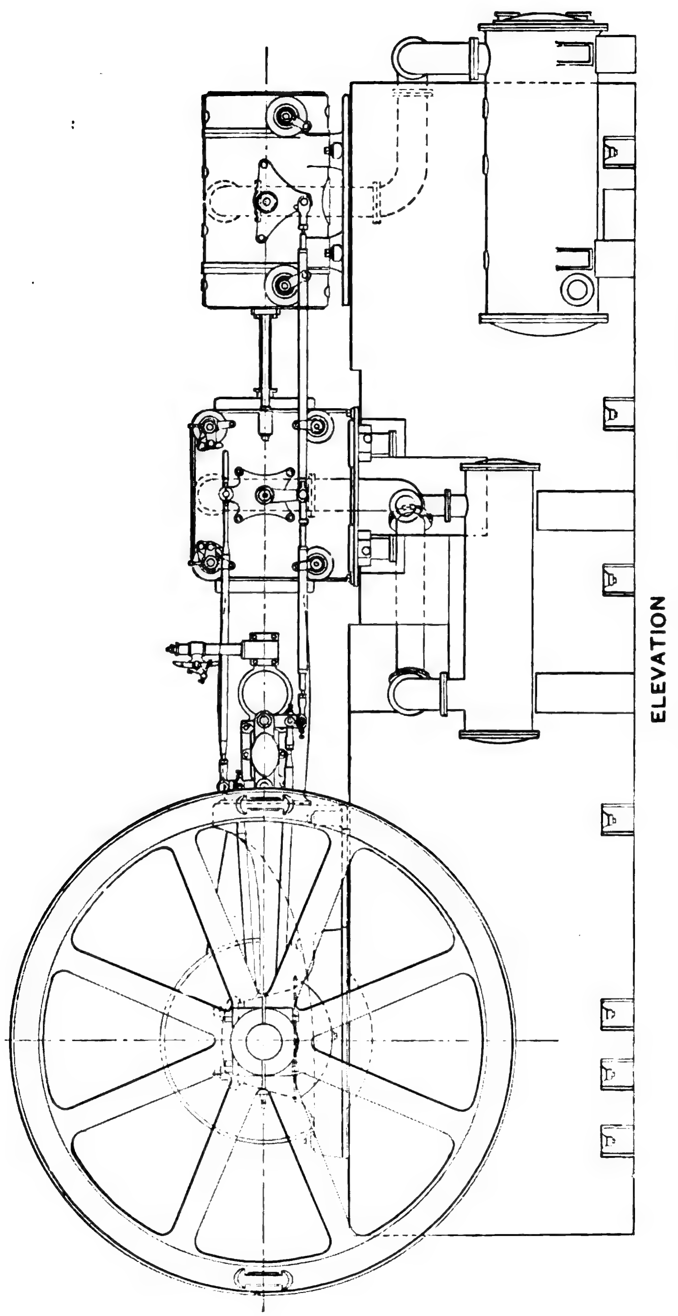


FIG. 20.—Allis-Chalmers Cross-Compound Corliss, Two-Stage Compressor.

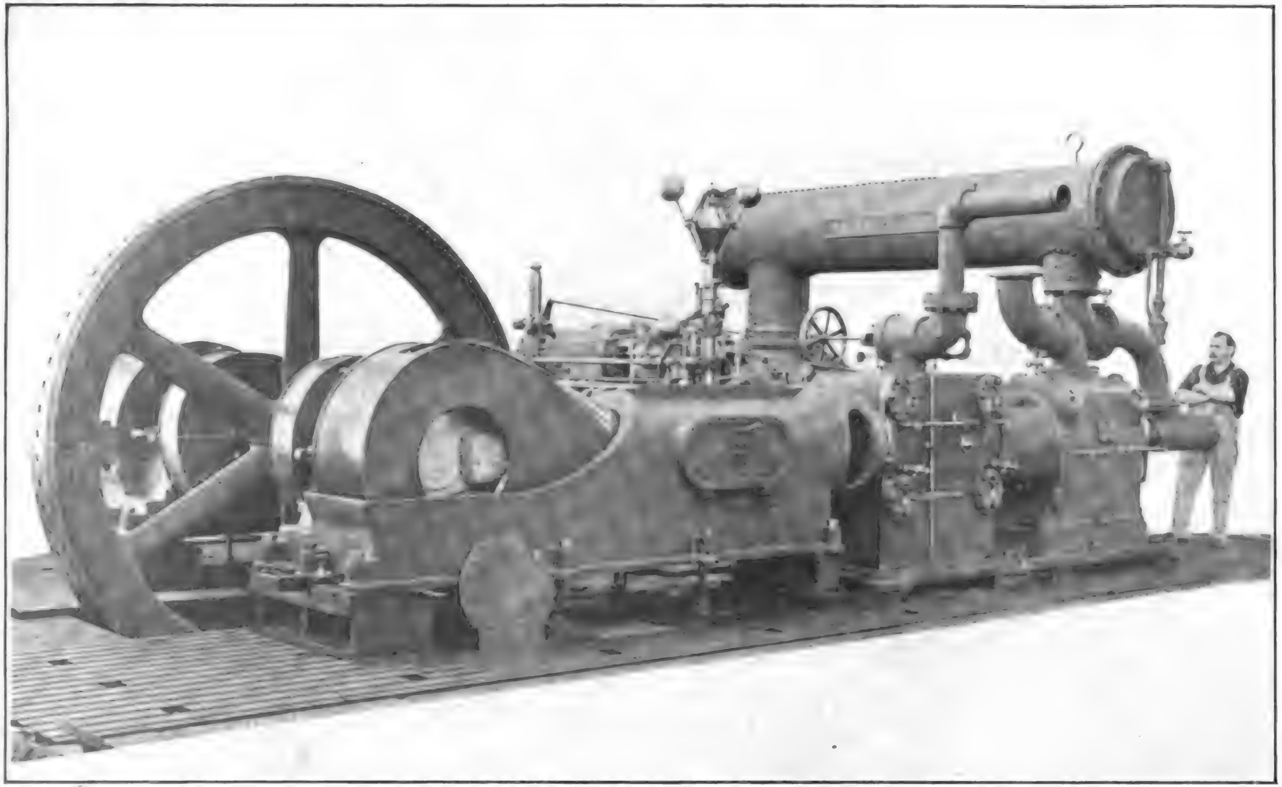
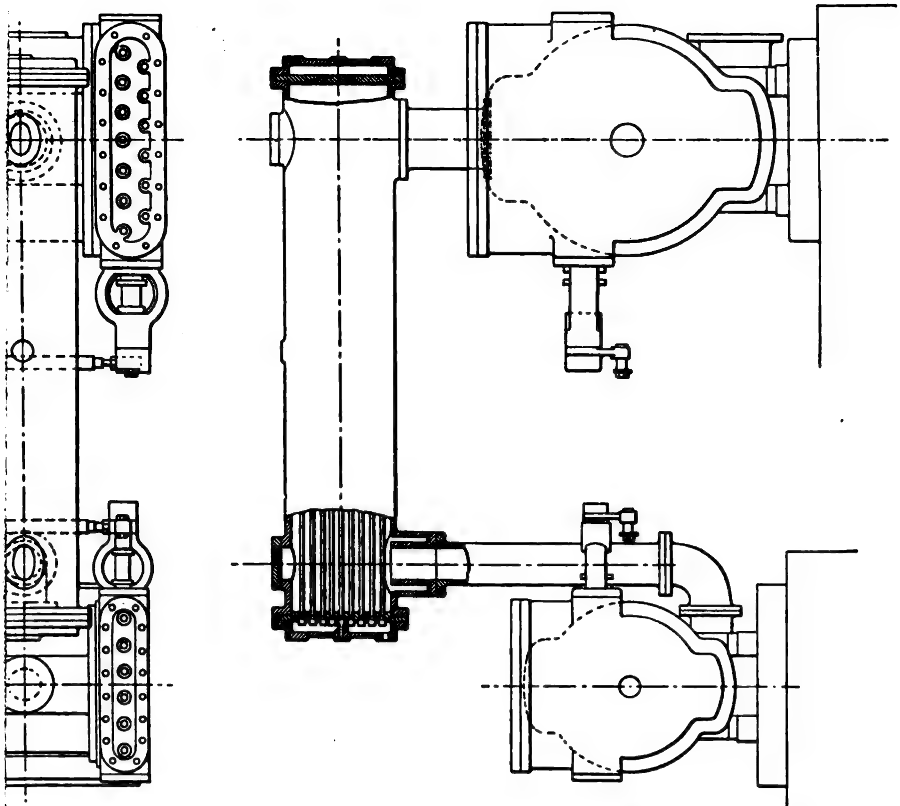


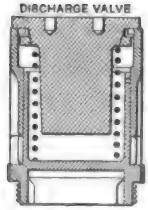
FIG. 21.—Ingersoll-Rand Cross-Compound, Two-Stage Compressor, Class "O."

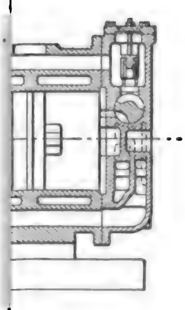
monly employed for machine drills, a saving in steam consumption can be realized. In elevated mountain regions, where so much mining is carried on, the advantages of stage compression are still greater than at sea-level, as is shown in Chapter XIII. The duplex form, with both steam and air ends compounded, exemplifies the highest type of compressor. There is no material increase in the number of moving parts, except valves; the greatest range of steam expansion is obtainable, because the work done in the air cylinders is more nearly equalized, and the compressor may be made self-regulating over its entire range of load. Thermodynamically, the efficiency of stage compression depends largely on the proper use of water-jackets for the cylinders, and the size and design of the intercooling apparatus between the air cylinders; a subject much better understood now than formerly. Stage compression is discussed in detail in Chapter VI.

Operation of Steam-driven Compressors. A steam-driven air compressor operates under peculiar conditions; appearing to work under a disadvantage which does not obtain in ordinary steam engines. This will be understood by inspecting the combined air and steam indicator cards of a simple straight-line compressor (Fig. 24). At the beginning of the stroke the air in front of the piston is at atmospheric pressure. As the piston advances the pressure at first increases slowly, while toward the end of the stroke it rises very rapidly. In other words, the resistance in the air cylinder varies from zero at the beginning of the stroke to its maximum near the end. The power developed in the steam cylinder, on the contrary, when working as usual with a cut-off, is in exactly the reverse order. The initial steam pressure may be even lower than the final air pressure, though the mean effective pressure in the steam cylinder is greater than the mean effective in the air cylinder, as shown by the diagram. For example, with an initial steam pressure of sixty pounds, air may be compressed to eighty pounds or more. This result is obtained by the use of heavy flywheels and reciprocating parts, for carrying the engine over its centers, storing up the surplus power in the early part of the stroke, and giving it out toward the end. It follows that there is a marked want



END ELEVATION





Compressor. (End elevation of air end;
s, with detail of discharge valve.)

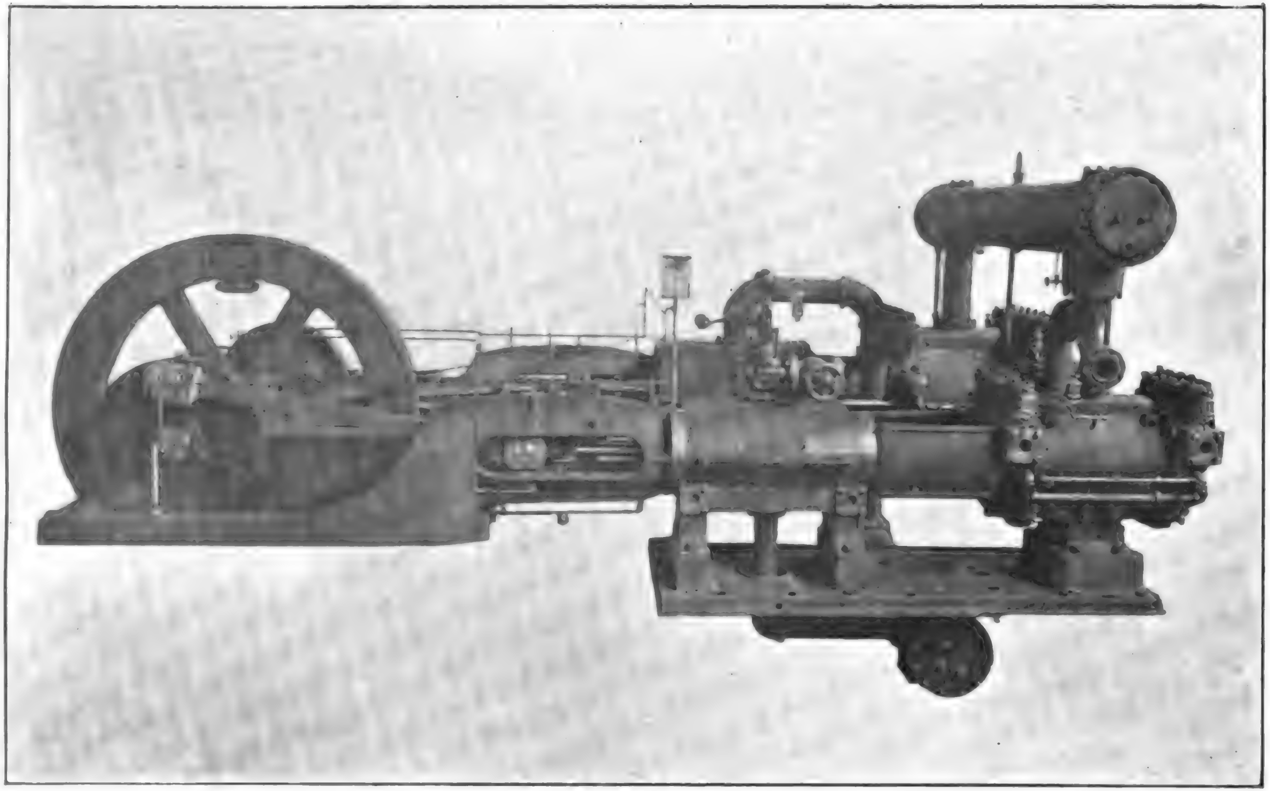


FIG. 22.—Laidlaw-Dunn-Gordon Duplex Cross-Compound Compressor, with Two-Stage Air Cylinders.

of smoothness in the running of compressors, which causes severe strains in the moving parts. This is specially noticeable in the simple straight-line type, which, when the air in the receiver is up to gauge pressure, will often be brought almost to a standstill and barely turn over the centers. It would thus appear that only a small ratio of expansion in the steam cylinder could be employed, and in fact some of the older forms of straight-line compressors took steam throughout nearly the entire stroke. But the difficulty is met, and greater economy made possible, by the inertia of the fly-wheels. The dimensions of the steam and air cylinders in simple compressors are proportioned for a cut-off of from $\frac{3}{8}$ to $\frac{1}{2}$ stroke.

In most of the simple straight-line compressors the steam cylinder is provided with an adjustable cut-off valve (Fig. 1). This valve *a* is composed of two parts and, moving on top of the main valve, controls ports in the latter through which steam is admitted to the main ports. It is operated by a separate eccentric on the fly-wheel shaft, and by means of the hand-wheel *b*, outside of the end of the valve chest, may readily be regulated without stopping the compressor, according to the varying pressure in the receiver. By manipulating this valve the compressor may be prevented from sticking on a dead center, notwithstanding considerable variations in receiver pressure.

A number of arrangements have been devised in the past to equalize the power and resistance, by varying with respect to one another the positions of the air and steam cylinders and their cranks. For example, in the earlier forms of the Burleigh, De la Vergne, and Ingersoll compressors, the cylinders, instead of being parallel to each other, were placed at 90°, with the cranks at 30°. In the old Rand and Waring, of 1876, the cylinders were set at 45°, the steam cylinder being of the oscillating pattern. The object of these and other similar devices was so to time the movements of the air and steam pistons that the power developed in the steam cylinder should be at its maximum when the air piston was just completing its stroke. But such constructions are deficient in strength and rigidity. They require heavier and more expensive

engine frames and foundations, and have not given satisfactory results.

In the duplex type, as already explained, the lack of equalization between power and resistance is minimized, the most favorable distribution and the highest degree of economy being attained in duplex stage compressors with compound steam cylinders.

Proportions of Cylinders. It is customary to build compressors with a short stroke, as this is conducive to economy in compression, as well as the attainment of a proper rotative speed. A short stroke is of special importance in simple straight-line compressors, because the power and resistance are more nearly equalized than

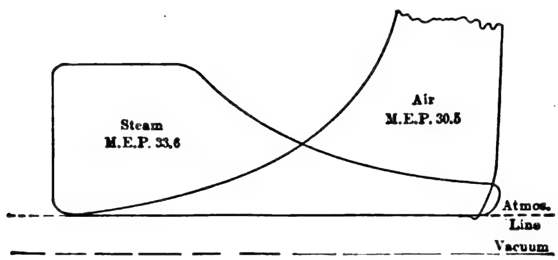


FIG. 24.—Combined Air and Steam Cards.

with a long stroke. The motion is less jerky and there is less liability of stopping on a center. With a long stroke, and relatively small diameter of cylinder, the piston would travel some distance under a constantly increasing resistance; then, after the discharge valves open, it would advance a considerable distance farther under a uniform resistance, while adding nothing to the amount of useful work. It should be noted, however, that the loss of capacity of the compressor due to piston clearance is less for a long than a short cylinder of the same diameter. In ordinary single-stage, slide-valve compressors the usual ratio of length of stroke to diameter of steam cylinder is 1⅛ to 1 or 1¼ to 1. In some makes, such as the older Rand compressors, the ratio was considerably greater, varying from 1½ or 1⅔ to 1. The length and diameter of steam cylinders in some recent designs are nearly equal. Quite different practice

prevails, however, in the design of duplex Corliss compressors. In these are found such variations in the proportions of steam cylinders as: 12″ × 30″, 14″ × 42″, 20″ × 42″, and 30″ × 60″.

The relative diameters of the air and steam cylinders depend obviously on the steam pressure carried and the air pressure to be produced. In mining operations there is usually but little variation in these conditions. For rock-drill work, the air pressure is generally from sixty to eighty pounds. Of late, however, the applications of compressed air for manufacturing purposes have so multiplied that some builders furnish compressors with steam and air cylinders of a great variety of proportions, for producing pressures of from ten to 120 pounds per square inch.

Compressors Driven by Water-Power. When available, water-power furnishes a cheap and convenient means of driving air compressors. Impulse or tangential wheels, such as the Pelton, Knight, or Risdon, are best adapted for this service, the wheel being mounted directly on the crank-shaft, as shown by Fig. 25. This cut is of a 16″ × 30″ compressor, built by the Risdon Iron Works for the Goleta Mining Co. It is driven by a sixteen-foot wheel under a head of 300 ft. Figs. 26 and 27 show plan and elevation of another compressor by the same makers. Plants similar to this are built by the Compressed Air Machinery Co., Ingersoll-Rand Co., and other makers. Since the power developed is uniform throughout the revolution of the wheel, water-driven compressors should be of the duplex type, in order to equalize the resistance as far as possible. The rim of the wheel is made extra heavy, to supply the place of a fly-wheel. This is illustrated by Fig. 28, of an Ingersoll-Rand compressor driven by a Pelton wheel.

To obtain the best efficiency, the peripheral velocity of an impulse wheel should be theoretically one-half the velocity of the jet of water from the nozzle. It follows that high heads of water involve correspondingly high peripheral velocities, and if the wheel be of small diameter a belt-drive would be required. But belting or gearing can generally be avoided, except when for any reason a turbine-wheel is adopted. Belt transmission is always disadvantageous, on account of the loss of power (say, eight to ten per cent.)

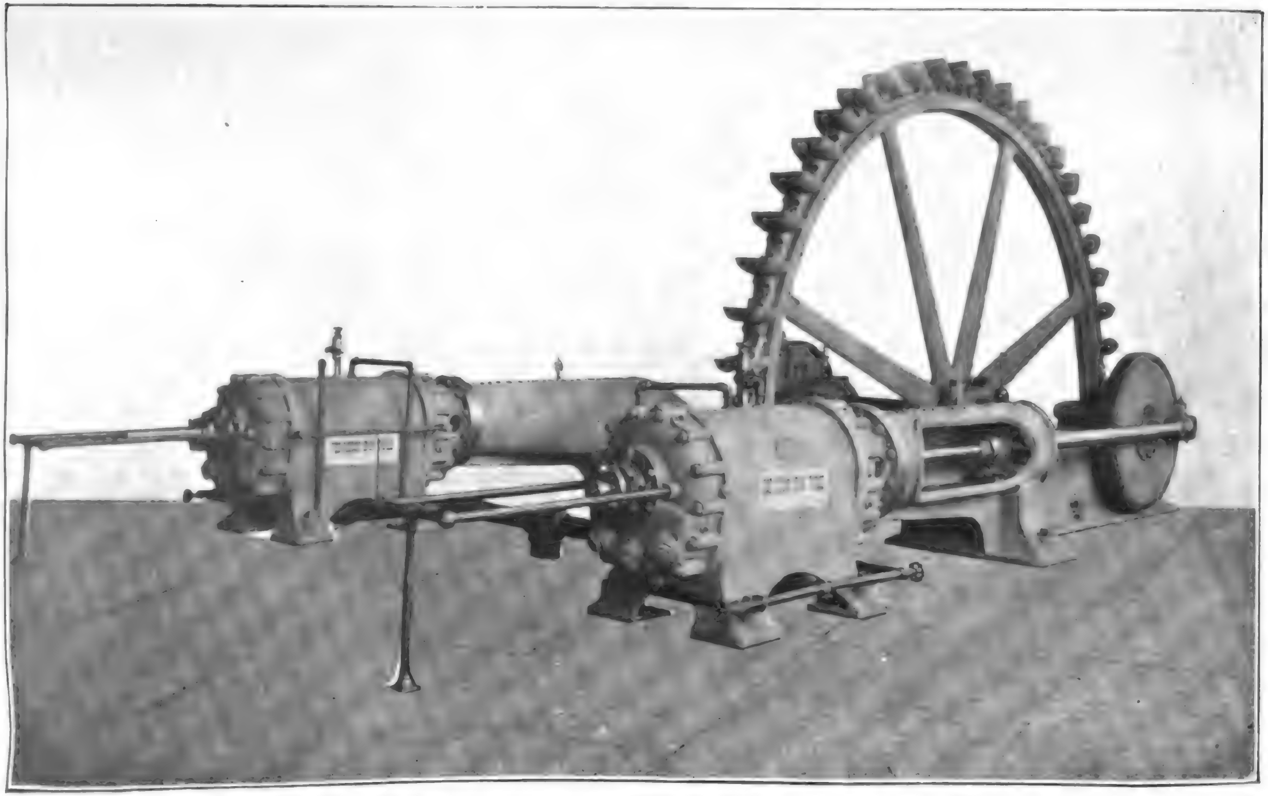


FIG. 25.—Duplex Compressor, with 16″ x 30″ cylinders, direct-connected to a 16-foot Risdon water-wheel. Built by Risdon Iron Works.

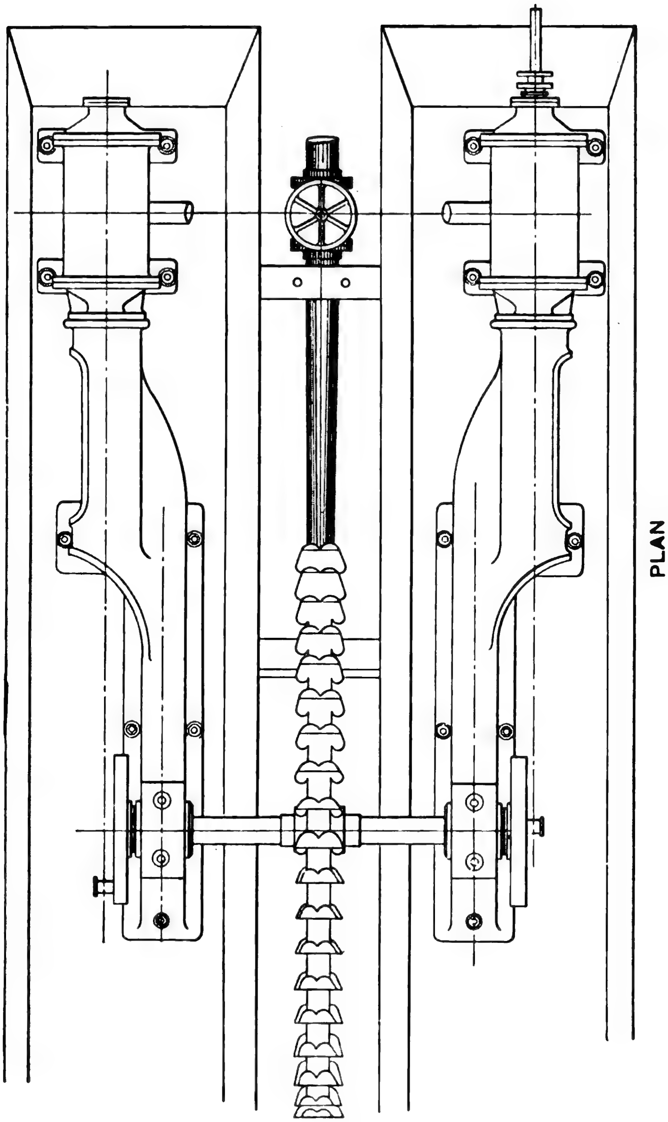
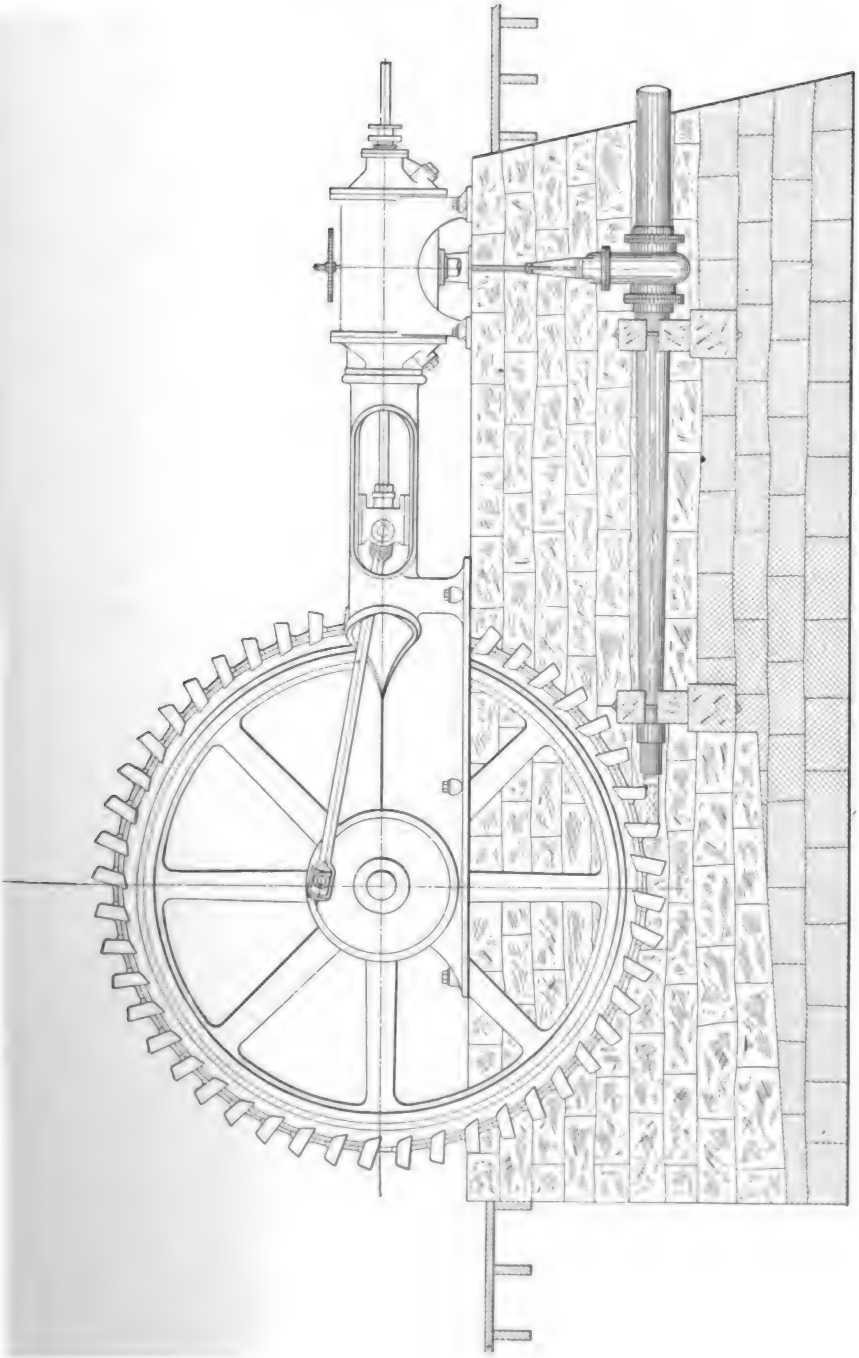


FIG. 26.—Water-Driven Duplex Compressor (Risdon Iron Works). PLAN.



ELEVATION

Fig. 27.—Water-Driven Duplex Compressor (Risdon Iron Works). Elevation.

and the cost of deterioration of the belting. Practically in all cases an impulse wheel can be made of large enough diameter to run at a peripheral speed which will insure economical use of the water, while still giving a sufficiently low rotative speed for direct-connected compressor cylinders. In accomplishing this with very heavy heads, water-wheels are sometimes made of great size.

Figs. 29 and 30 illustrate a well-known and interesting plant at the North Star Mine, Grass Valley, Cal., where a thirty-foot Pelton wheel drives a 300-horse-power, four-cylinder, two-stage compressor. The wheel makes sixty-five revolutions per minute under a head of 775 ft., with a single $1\frac{3}{4}$ inch nozzle. The cylinders are single-acting (to obtain more efficient cooling of the air and make it easier to detect piston leakage) and measure 30 and $18\frac{1}{2}$ inches $\times$ 30 inch stroke. To avoid building excessively high foundations, as would otherwise be necessary for a wheel of this size, and also to furnish a substantial support for the gearing, the cylinders are set on angular frames at 30° to the horizontal. In the lower left-hand corner of Fig. 29, the intercooler is shown, submerged in the tail-race, a feature of the plant of no small advantage in producing a thorough cooling of the air without expense. The spur-wheel on the main shaft, with its accompanying pinion, is provided for operating the compressor, if necessary at any time, by a synchronous electric motor (shown in outline to the left, in Fig. 30). At the extreme left of Fig. 30 is a small auxiliary water-wheel, for starting the motor and putting it in synchronism.*

At the Morning Mine, near Mullan, Idaho, is another large water-driven two-stage compressor, of an entirely different design. There are four cylinders, a high- and a low-pressure being set tandem on each side of a set of three Pelton wheels, mounted on the crank-shaft. A large volume of water, under a head of 140 feet, is delivered through 8,000 feet of flume and 400 feet of pressure pipe, driving two 12-foot wheels. Two other streams, piped respec-

* The bevel gears driven from the spur and pinion do not form part of this plant. They were used at one time to drive another compressor. For a more detailed description of this compressor, the illustrations of which were kindly sent to the writer by its designer, Mr. E. A. Rix, see *American Machinist*, November 10th, 1898, p. 831.

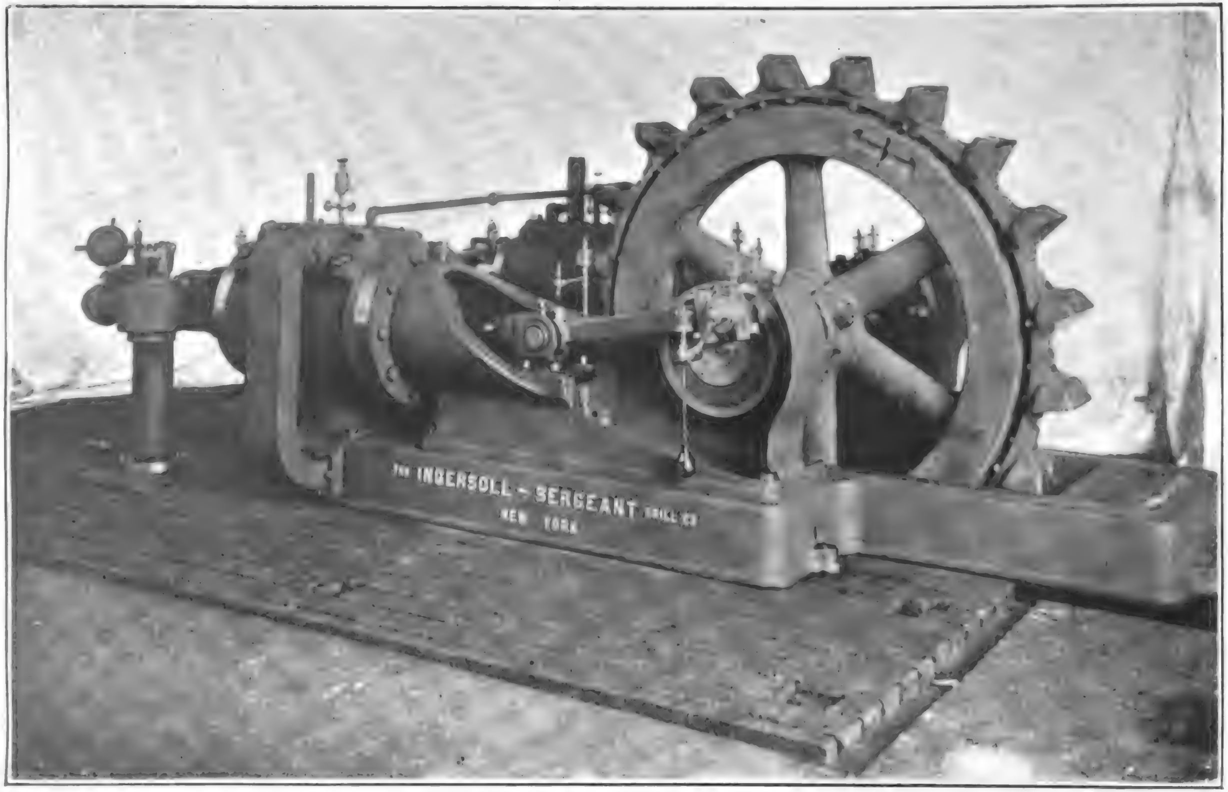


FIG. 28.—Ingersoll-Rand Compressor, Driven by a Pelton Water Wheel.

tively $1\frac{1}{2}$ and 1 mile, produce heads of 1,140 and 1,420 feet. These drive a 33-foot Pelton wheel (probably the largest in the world), placed on the compressor crank-shaft between the smaller wheels. The central wheel is driven by separate jets from the high-pressure lines, and on account of their difference in head, the diameter adopted for this wheel is a mean between the diameters theoretically necessary for obtaining a peripheral velocity properly proportioned

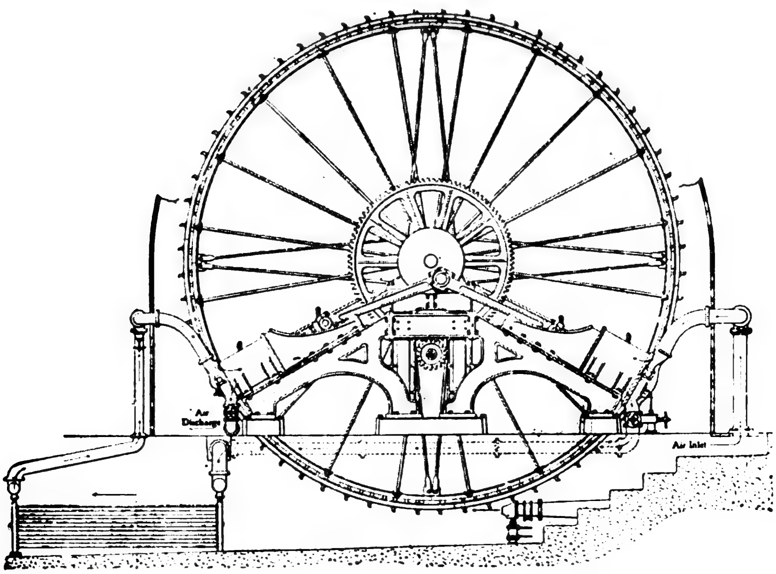


FIG. 29.—Water-Driven Compressor at the North Star Gold Mine, California. (Side Elevation.)

to each head. An actual mean peripheral speed of 8,000 feet per minute is attained. To control the water under these great heads, which correspond to pressures of about 490 and 610 pounds per square inch, slow-acting gate valves are provided, with by-passes for use in starting and stopping. The nozzles are arranged to be deflected clear of the wheel, in case it should be necessary to stop the compressor quickly.

Each pair of cylinders are $33\frac{1}{2}$ and 18 inches respectively $\times$ 42-

inch stroke, working at a piston speed of 560 feet. The low-pressure cylinders compress to about 30 pounds, the high-pressure to 90 pounds. Inter- and after-coolers are placed in the tail-race of the smaller wheels. A positive valve-motion is employed for both inlet and discharge valves, which are of the Corliss type. On each side, parallel to the center line of the compressor and geared to the crank-shaft, is a long shaft. Geared to the latter in turn are

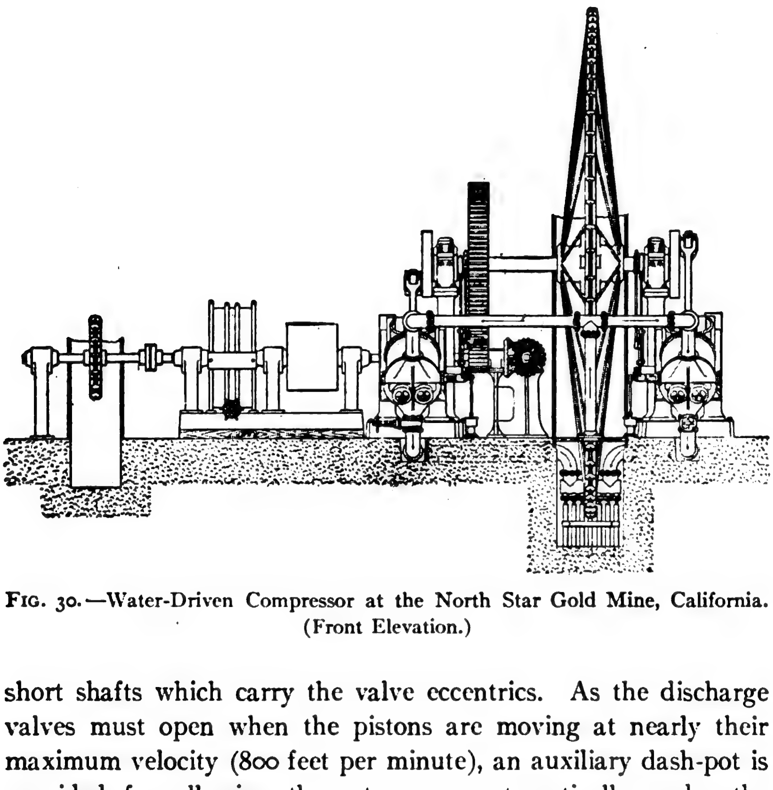


FIG. 30.—Water-Driven Compressor at the North Star Gold Mine, California. (Front Elevation.)

short shafts which carry the valve eccentrics. As the discharge valves must open when the pistons are moving at nearly their maximum velocity (800 feet per minute), an auxiliary dash-pot is provided for allowing them to open automatically under the cylinder pressure, the positive eccentric motion closing them.

Indicator cards from this compressor show it to be highly efficient. An average of a number of cards gives mean pressures of: low-pressure cylinder, 17.86 pounds; high-pressure, 41.14 pounds;

combined, 30.46 pounds. The mean theoretical adiabatic and isothermal pressures, corresponding to the combined mean are, respectively, 36.94 and 28.5 pounds. During the tests the observed temperatures were: cooling water, 38°; air at discharge from low-pressure cylinder, 135°; air at high-pressure inlet, 46°; high-pressure discharge, 140°; on leaving the after-cooler, 62°. Mean atmospheric temperature, 55° and of the cooling water 38°.*

Provided there is a sufficient volume of water, impulse wheels may be used with quite low heads, by introducing multiple nozzles, directed tangentially at two or more points of the periphery of the wheel. To prevent the water from splashing over the compressor, the wheel is enclosed in a tight wooden or iron casing. The force of the water may be regulated by an ordinary gate-valve; but if the head be great it is always desirable to use a special slow-moving gate (as noted above), to avoid danger of rupturing the pressure pipe in case the compressor is suddenly stopped. Turbines are obviously not so well adapted for operating compressors as the impulse wheels. A method of compressing air by the direct action of falling water is described in Chapter XV.

Belt-Driven and Geared Compressors. These are often convenient, and are furnished in a number of styles and sizes by compressor-builders. The fly-wheel is replaced by a large belt-wheel, with an extra heavy rim to give it sufficient weight, Fig. 31. The compressor illustrated is a recent design of the Ingersoll-Rand Co. In Fig. 32 is shown one of the older machines by the same makers. Power may be derived from an engine already installed for other purposes, or from a water-wheel or electric motor. Since electric transmission of power has come into general use in mining regions, a belt-drive from a motor is frequently advantageous when there is sufficient floor space. Some of the compressor-builders have introduced a "silent-chain" drive, for use when it is desired to place the motor close to the compressor and on the same bed-frame, and at the same time avoid the use of gearing. It has a high efficiency (about ninety-five per cent.) and may be employed for transmitting up to, say, 200 horse-power.

* This plant, described in *American Machinist*, September 26th, 1901, was, like that at the North Star mine, designed by Mr. E. A. Rix.

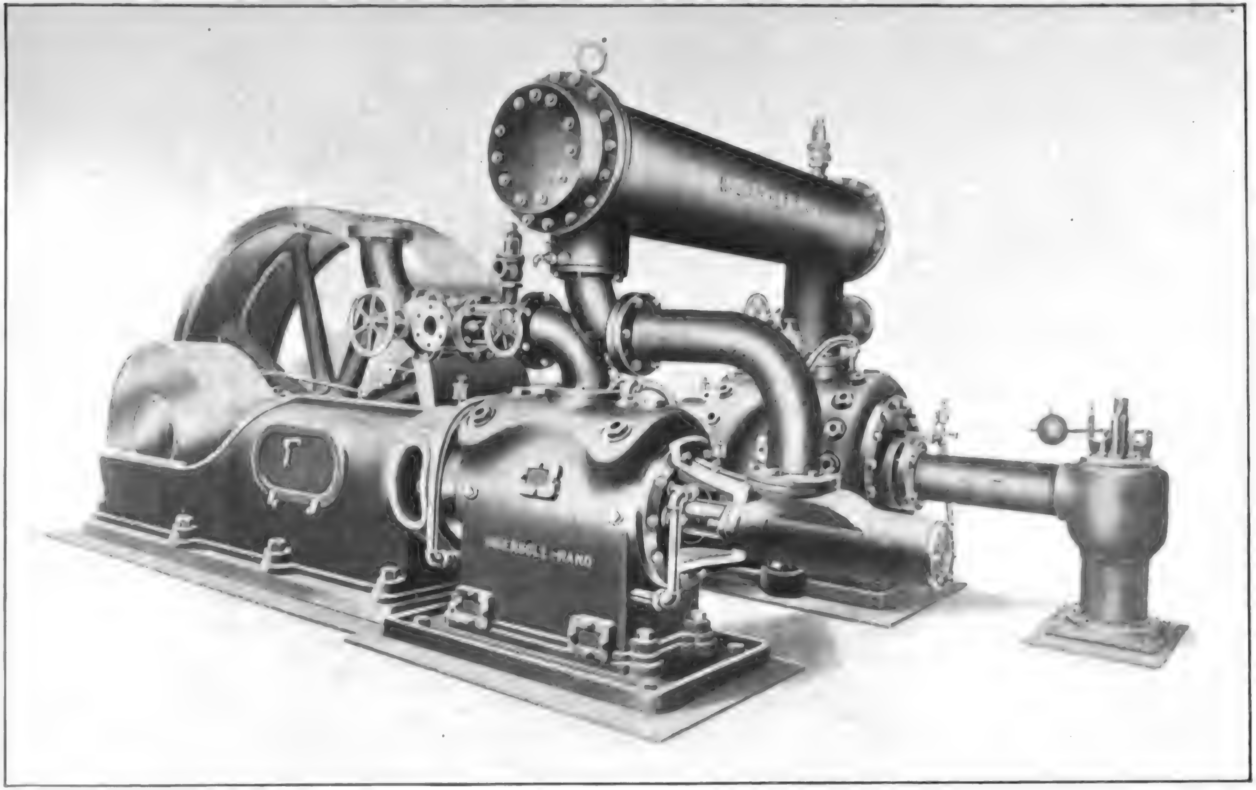


FIG. 31.—Ingersoll-Rand Duplex, Two-Stage, Belt-Driven Compressor.

Although a belt-drive is preferable to gearing, at least for a compressor erected on the surface, geared electric-driven sets are sometimes used, a spur-gear on the crank-shaft engaging with a pinion on the armature. Single-reduction gearing will generally answer. This design has been adopted even for large plants, as, for example, at a recent two-stage installation of the Compañia de Peñoles, Mexico. By giving sufficient diameter and weight to the spur-

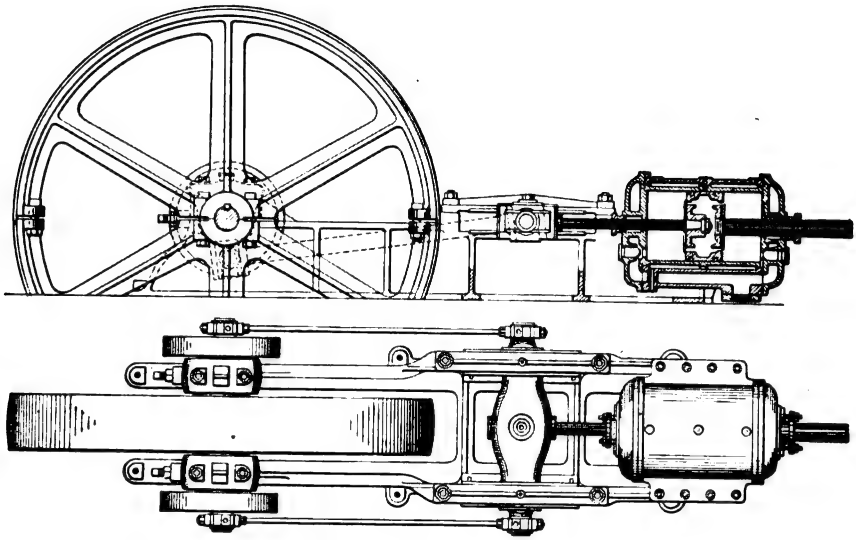


FIG. 32.—Ingersoll-Rand Straight-Line, Belt-Driven Compressor.

wheel, it not only produces the low piston speed necessary, but serves also as a fly-wheel. Raw-hide pinions are desirable to reduce the noise of the gearing. Induction motors are suitable for such service, as they are capable of running economically under wide variations of load. It may be added that the small, high-speed Christensen compressor is well adapted for gearing directly to a motor, thus forming a very compact machine for uses where lightness or portability is essential. Fig. 33 shows a longitudinal section through the low-pressure cylinder of a recent design of a direct-connected, electrically driven, duplex compressor, built by

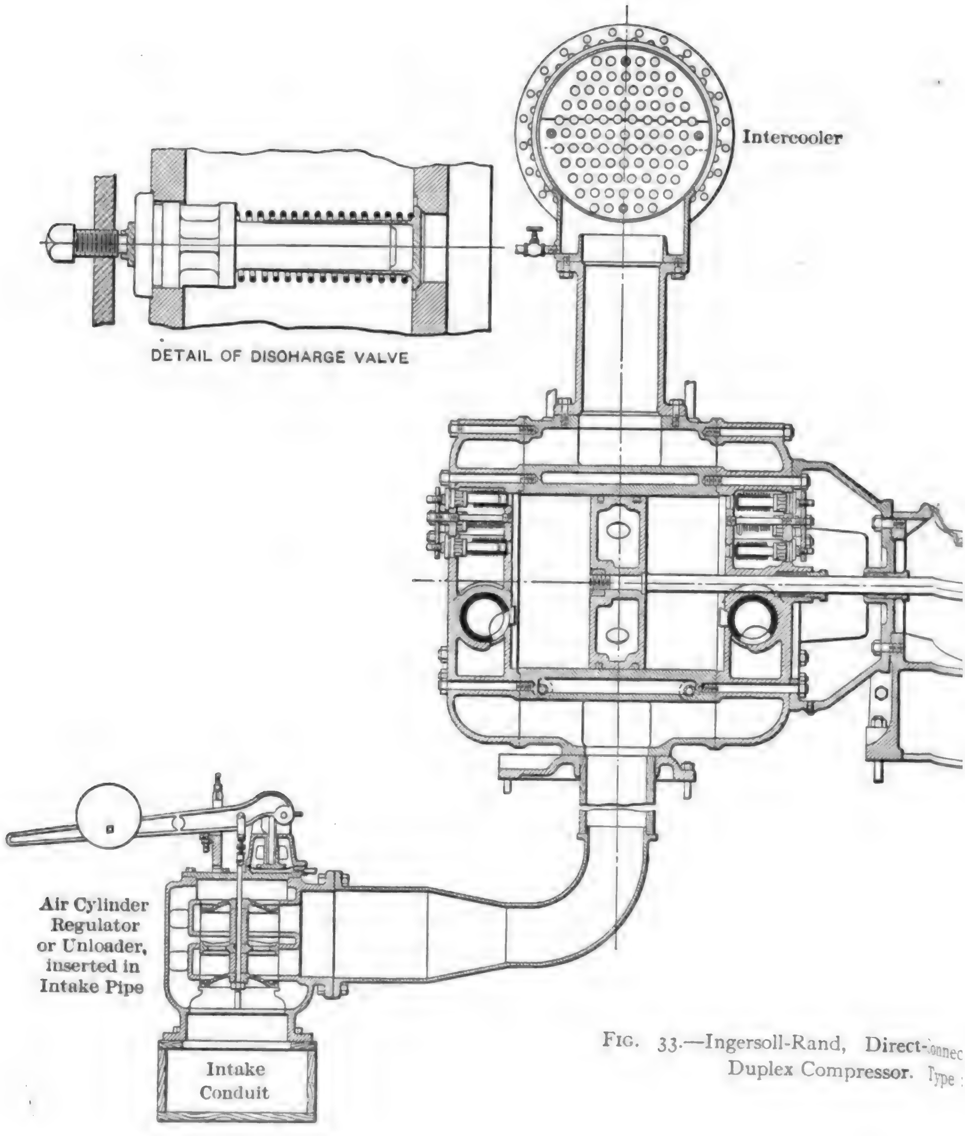
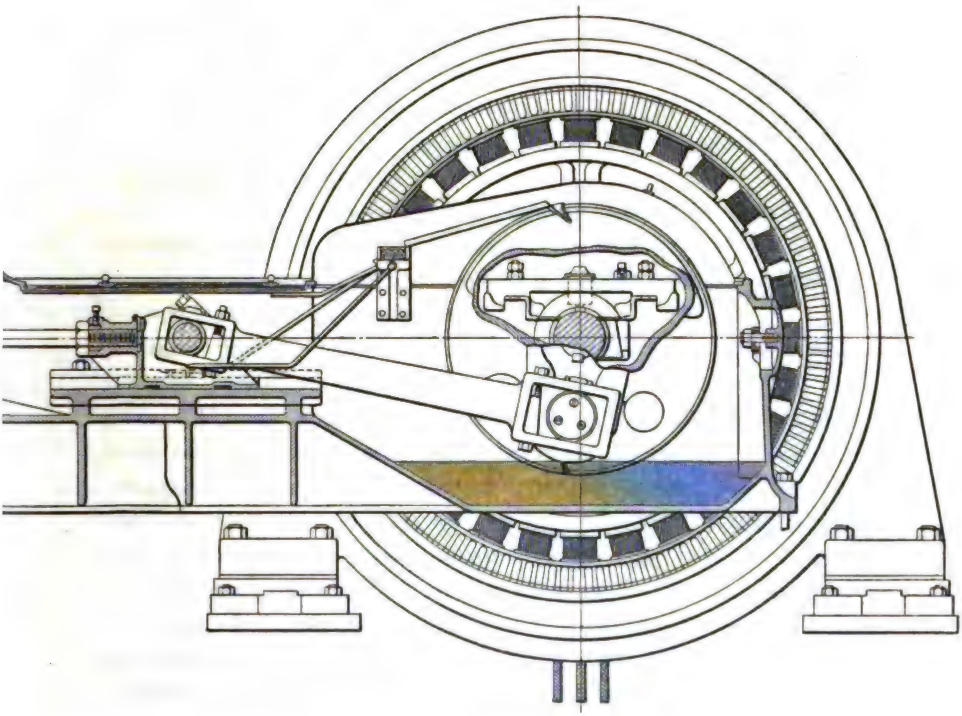


FIG. 33.—Ingersoll-Rand, Direct-connec
Duplex Compressor. Type



nnected, Electrically-Driven, "Imperial"
ype 10. Longitudinal Section.

the Ingersoll-Rand Co. One of the features of the compressor is that the frame is inclosed and all parts are self-oiling, except the piston and cylinder. The crank-pit contains the oil supply. The oil is picked up by the edge of the crank disc, and taken off at the top by a scraper. Part of it goes to a distributing tank, from which small pipes lead to the cross-head and guides. The main bearing and crank-pin are oiled direct from the scraper by a projecting trough on each side. For the bearing the oil is led to a channel in the cap, whence it passes through a series of holes drilled through the bearing liner. Some of this oil goes to the collar of the eccentric hub, from which it is carried to the face of the eccentric through two holes. As the eccentric is closed, the surplus oil is returned to the crank-pit. Thus the oil-feed is proportioned to the speed of the compressor, ceasing entirely when the compressor is stopped.*

These direct-connected compressors are driven by direct-current induction or synchronous motors, the rotors of which are of large diameter, as shown, to produce a proper relation between the peripheral and rotative speeds.

Under proper conditions an electric-driven compressor may be erected underground, near the point of application of the air power. Though some loss is inevitable in converting electric energy into compressed-air power, this may be offset in some circumstances by considerations of convenience of installation. When the electricity is generated by water-power in large quantities, as in many Western mining districts, the cost per horsepower compares favorably with that of steam-driven compressors.

NOTE. It is unnecessary, and in fact hardly practicable, in a book that is not intended to be a trade publication, to describe separately and in detail all the numerous makes of air compressors. In the foregoing chapter some of the well-known compressors are instanced, for the purpose of illustrating various features of design. The same remark may be made with regard to the descriptions of air valves, etc., in Chapters VII, VIII, and IX. It must not be under-

* Similar self-oiling systems are applied to the compressors shown in Figs. 15, 21, and 31.

stood, however, that the compressors specifically referred to in this book are considered the only good ones, nor that the author, by omitting to mention and to insert cuts of all compressors, desires thereby to discriminate against those that are perhaps less well known only because they are the product of recently established builders. Most of the compressors made in Europe, including many excellent machines, are omitted altogether, though references to interesting features of the valve-motions of some of them will be found under the appropriate heads.

An alphabetical list, which, while incomplete, comprises the names of most of the American compressor-builders, is given below.

- Allis-Chalmers Co.
- American Air Compressor Works.
- Chicago Pneumatic Tool Co. (Franklin compressor).
- Christensen Motor-Driven Compressor (Allis-Chalmers Co.).
- Clayton Air Compressor Works.
- Compressed Air Machinery Co.
- Franklin Iron Works.
- Heron & Bury Manufacturing Co.
- Ingersoll-Rand Co.
- Knowles Steam Pump Works.
- Laidlaw-Dunn-Gordon Co.
- Leyner, J. Geo., Manufacturing Co.
- McKiernan Drill Co.
- New York Air Compressor Co.
- Nordberg Manufacturing Co.
- Norwalk Iron Works Co.
- Rix Compressor and Drill Co.
- Sullivan Machinery Co.
- Vulcan Iron Works.

CHAPTER III

THE COMPRESSION OF AIR

In the production and use of compressed air occur serious losses, which to a large extent are unavoidable. Even in the best compressors the efficiency, or ratio of the force stored up in the compressed air to the work which has been expended in compressing it, rarely exceeds seventy-five per cent. and often falls below sixty per cent. To understand the causes of these losses it will be necessary to study the principles involved in the operation of compressing air. This study is advisable, also, before proceeding to a description of the air end of the compressor. Several definitions may first be given:

"Free air" is a term commonly used in dealing with the subject of air compression. It is simply air at normal atmospheric pressure, as taken into the cylinder of the compressor. But since atmospheric pressure varies with the altitude above sea-level, and with the barometric reading at any particular time or place, it follows that the expression "free air" has no precise general signification, with respect to the pressure, volume, and temperature of the air. At sea-level it is in reality "compressed air," at the normal atmospheric pressure of 14.7 pounds per square inch. As commonly employed the term means air at sea-level pressure, and at a temperature of 60° Fah.

The **absolute pressure** of air is measured from zero, and is equal to the assumed (or observed) atmospheric pressure plus gauge pressure; ordinary gauges registering pressures in pounds per square inch above atmospheric pressure.

Absolute temperature is the temperature as measured from the "absolute zero" point, which is 491.4° F. below the freezing-point of water, or say 459° below zero Fahrenheit. For example, 60° F.

of thermometric temperature is equivalent to an absolute temperature of 459° + 60° = 519° F.

There are two fundamental laws governing the behavior of a perfect gas, when undergoing compression, which for all practical purposes are applicable also to atmospheric air. In discussing the problems of air compression, all the relations existing between volume, pressure, and temperature may be expressed in accordance with these laws. The first law (Boyle's) is: At constant temperature the volume occupied by a given weight of air varies inversely as the pressure. This condition is expressed by the equation:

$$P\,V = P'\,V' = \text{constant; or } \frac{P'}{P} = \frac{V}{V'}; \text{ in which}$$

V = the volume of the given weight of air (or gas) at the freezing-point and at a pressure P (V usually being taken as the volume in cubic feet occupied by one pound of air); V' = the volume of the same weight of air at the same temperature and at any pressure, P' (the pressures being absolute pressures).

For example, to compress a quantity of atmospheric air at constant temperature to 0.147 of its original volume (the atmospheric pressure being 14.7 pounds), requires a pressure of 100 pounds per square inch; when compressed to 0.074 of its original volume, the pressure required is 200 pounds, and so on.

TABLE I

Temperature Degrees Fah.	Weight of one Cubic Foot in Pounds.	Volume of one Pound in Cubic Feet.	Temperature, Degrees Fah.	Weight of one Cubic Foot in Pounds.	Volume of one Pound in Cubic Feet.
0	.0863	11.582	110	.0697	14.345
10	.0845	11.834	120	.0685	14.596
20	.0827	12.085	130	.0674	14.847
30	.0811	12.336	140	.0662	15.098
32	.0807	12.386	150	.0651	15.350
40	.0794	12.587	160	.0641	15.601
50	.0779	12.838	170	.0631	15.852
60	.0764	13.089	180	.0621	16.103
62	.0761	13.141	190	.0612	16.354
70	.0750	13.340	200	.0602	16.605
80	.0736	13.592	210	.0593	16.856
90	.0722	13.843	212	.0591	16.907
100	.0710	14.094			

Table I * shows the weight and volume of dry air, at temperatures from 0° to 212° F., and at atmospheric pressure.

The production and use of compressed air, if governed solely by the law stated above, would be a simple matter. But during compression heat is generated, and when the air is allowed to re-expand to its original volume this heat is given up. Provided there is no transference of heat, the internal work, manifested by the increase of temperature, is independent of the time occupied by the compression. This condition is expressed by the second law, that of Charles and Gay-Lussac, *viz.*: When under constant pressure, the volume of a gas expands or contracts for each degree rise or fall of temperature, from freezing to boiling, by a constant fraction of the volume which it occupied at the freezing-point. Stated in another way, the volume of a gas under constant pressure is nearly proportional to the absolute temperature. The equation may be written: $V' = V\ (1 + at^\circ)$. The complete relations between pressure, volume, and temperature are expressed by the equation: $P'V' = PV\ (1 + at^\circ)$, in which P' and V' represent the pressure and volume of a given weight of air (or gas) at t° Fah. above the freezing-point, P and V the pressure and volume of the same quantity of air at the freezing-point, and a the coefficient of expansion of air, which is practically constant and is very nearly $\frac{1}{491}$ on the Fahrenheit scale. Hence, for a rise in temperature of 1° F., the volume of the air increases by $\frac{1}{491}$ of the volume occupied at the freezing-point, under the same pressure, 491° F. being the absolute temperature below freezing.

The practical application of this law is that the development of heat reacts upon the air under compression, and increases the pressure due merely to the reduction in volume. By cooling the compressed air to its original temperature the pressure would be reduced to the normal amount, according to the first law. That is, the heat produced by the compression of a given volume of air corresponds in degree to the cold resulting from the re-expansion of the same quantity of air to its original volume and pressure. It is evident that this property of air has an important application in the production and use of compressed air.

* From D. K. Clark and Appleton's "Applied Mechanics."

Two other statements may be deduced from what precedes: 1. Under constant pressure the volume of air varies directly as the absolute temperature; 2. The volume being constant, the absolute pressure varies directly as the absolute temperature.

The first of these statements is expressed thus:

$$\frac{V}{t} = \frac{V'}{t'} = \text{constant},$$

in which t and t' are absolute temperatures; whence, from Boyle's law:

$$P\frac{V}{t} = P'\frac{V'}{t'} = \text{constant}.$$

For convenience, this constant is commonly denoted by R, and the general equation is written $PV = R\,t$, or $\frac{PV}{t} = R$.

The value of R is found as follows: Since for a given weight of gas or air the density, D, is inversely proportionate to the volume, $V = \frac{1}{.08073}$, .08073 being the weight in pounds of one cubic foot of dry air, at sea-level pressure (14.7 lbs.) and 32° F. The normal atmospheric pressure per square foot = 14.7 × 144 = 2,116.8 lbs. If, therefore, one cubic foot be expanded by the application of heat to a volume of two cubic feet, the work done against atmospheric pressure, per pound of air, will be $\frac{2116.8 \times 1}{.08073} = 26{,}220$ foot-pounds. To double the volume, according to Boyle's law, would require the expenditure of 491.4° F. of heat; whence, in raising the temperature 1° F., the external work done by expansion is:

$$\frac{PV}{t} = \frac{26220}{491.4} = 53.37 = R.$$

The heat generated during compression and corresponding to different pressures is shown in Table II, the volume at normal atmospheric pressure being 1, at a temperature of 60° Fah.

From this table it is seen that the *rate* of increase of temperature is not uniform, but diminishes as the pressure rises. Thus, from 1 to 2 atmospheres the increase is 115.8°; from 2 to 3, 79.3°; from

TABLE II

Pressure in Atmospheres.	Absolute Pressures, Pounds per Square Inch above Vacuum.	Volumes in Cubic Feet, Adiabatic Compression.	Final Temperatures, Degrees Fah.	Corresponding Increases of Temperature.
1.00	14.70	1.000	60.0	00.0
1.25	18.37	0.854	94.8	34.8
1.50	22.05	0.750	124.9	64.9
2.00	29.40	0.612	175.8	115.8
2.50	36.70	0.522	218.3	158.3
3.00	44.10	0.459	255.1	195.1
3.50	51.40	0.411	287.8	227.8
4.00	58.80	0.374	317.4	257.4
5.00	73.50	0.319	369.4	309.4
6.00	88.20	0.281	414.5	354.5
7.00	102.90	0.252	454.5	394.5
8.00	117.60	0.229	490.6	430.6
9.00	132.30	0.211	523.7	463.4
10.00	147.00	0.195	554.0	494.0
15.00	220.50	0.147	681.0	621.0

3 to 4 atmospheres, 62.3°, etc. The quantity of heat developed during compression may be calculated by the following formula: *

$$Q = \frac{R \times t}{J} \times \text{Nap. log. } \frac{V'}{V}, \text{ in which}$$

Q = quantity of heat in thermal units.

R = constant = 96.037 (French unit) or 53.37 (English unit).

t = absolute final temperature in degrees, corresponding to V′ (centigrade scale for French and Fahrenheit for English units).

J = value of one thermal unit = 1,400 foot-pounds (or 778 foot-pounds if English units be used).

V and V′ = volumes of air in cubic feet, at beginning and end of compression.

As the rise in temperature due to compression is proportional to the ratio of the final absolute pressure to the initial absolute pressure, the quantity of heat generated during compression to any given pressure, and the consequent work done, is greater at high altitudes than at sea-level.

The above conclusions are illustrated by the diagram, Fig.

* Zahner, "Transmission of Power by Compressed Air," p. 109.

34.* It is, in reality, two diagrams, combined to save space. *First*, beginning at the lower left-hand corner, and curving upward, are the adiabatic and isothermal compression lines. Their intersections with the horizontal and vertical lines give the volumes of the unit of air when subjected to any given pressure, by reading the figures at the top, and right- or left-hand margin of the diagram. The initial volume is taken as 1, and the spaces between the vertical lines are each one-tenth. The resulting volume is independent of the initial temperature of the air. The corresponding pressure may be read in terms of either gauge or atmospheric pressure. *Second*, beginning at the lower right-hand corner of the diagram, and rising toward the left, are the lines of temperature, the assumed initial temperatures being 0°, 60° and 100° F.

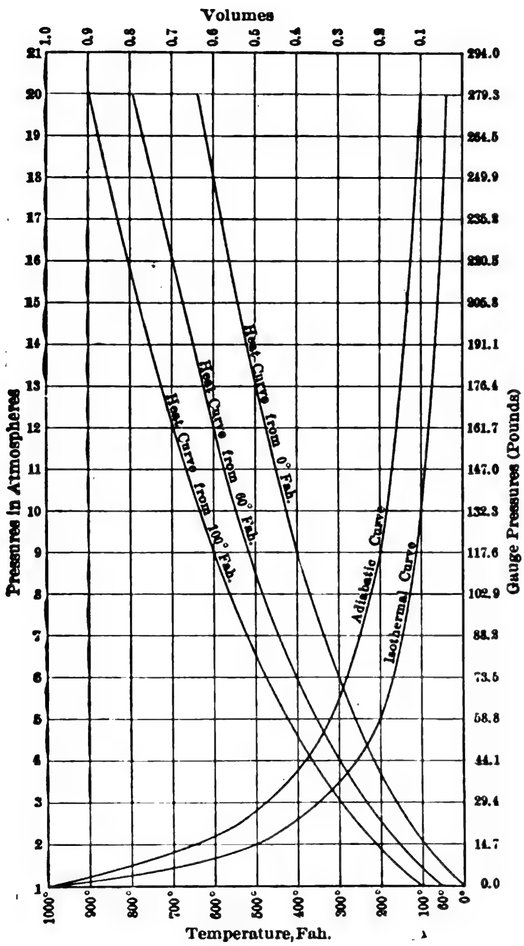


FIG. 34.

* Taken from "Compressed Air Production," by W. L. Saunders, several slight corrections having been made in the adiabatic and isothermal lines.

The temperature corresponding to any given pressure is read on the lower margin. It should be observed that these temperature curves are those of adiabatic compression.

It follows from the results obtained above that if the temperature of the air be allowed to rise during compression an increase of work ensues.

Isothermal and Adiabatic Compression. In accordance with the laws already stated, air may be compressed in two ways:

Isothermal Compression.—The temperature is kept constant during compression, the heat generated being abstracted as fast as it is developed. In this case the pressure of the air varies according to the equation $P V = P'V'$, or $\frac{P'}{P} = \frac{V}{V'}$, and such compression may be called isothermal; that is, the compression curve of an indicator diagram would be an isothermal curve.

Adiabatic Compression.—The temperature may be allowed to rise unchecked during the period of compression, as it will when there is no transference of heat, either by radiation or cooling devices. The rise in temperature increases the pressure that would be due to reduction of volume only. In other words, the pressure rises faster than the volume diminishes, and $\frac{P'}{P}$ is no longer equal to, but is greater than $\frac{V}{V'}$.

This relation is determined by a consideration of the specific heats of air at constant pressure and at constant volume. The specific heat of any gas or vapor at constant pressure, C_p, is the quantity of heat (in terms of heat units) required to raise the temperature of one pound of the gas 1° F., the pressure being unchanged. The specific heat at constant volume, C_v, is similarly the quantity of heat required to raise the temperature of the gas 1° F., the volume being unchanged. Regnault's experiments have shown that for air $C_p = 0.2375$ and $C_v = 0.1689$. Of these C_p is the greater, because external work is done during a change of temperature, if the pressure be constant and the air free to expand; while, under constant volume, no work is done upon external

resistances. When, as in adiabatic compression, the heat generated reacts on the air under compression and increases the value of $\frac{P'}{P}$, to maintain the equation, $\frac{V}{V'}$ must be increased by an amount equivalent to the external work performed. The specific heats may be expressed in heat units, as above; or, by multiplying them by the mechanical equivalent of a heat unit (778 foot-pounds = J), they are given in terms of foot-pounds and are then denoted by K; that is,

$$J\,C_p = K_p \text{ and } J\,C_v = K_v.$$

Since $K_p = C_p \times 778 = 184.77$, and $K_v = C_v \times 778 = 131.4$, the ratio of these quantities gives:

$$\frac{K_p}{K_v} = \frac{184.77}{131.4} = 1.406.$$

This ratio is commonly denoted by n, and is the exponent of the power to which $\frac{V}{V'}$ must be raised to make it equal to $\frac{P'}{P}$.* It is evident that n may also be expressed simply as equal to the ratio of the specific heat at constant pressure to the specific heat at constant volume:

$$\frac{C_p}{C_v} = \frac{0.2375}{0.1689} = 1.406 = n.$$

The general equation for adiabatic compression is therefore:

$$P\,V^n = P'\,V'^n \text{ or } \frac{P'}{P} = \left(\frac{V}{V'}\right)^{n\,=\,1.406}$$

Work of Compressors without Clearance. Isothermal Compression. The work done by a compressor without clearance, and using isothermal compression, is represented by the area under the compression curve (Fig. 35). Let A B be an isothermal curve, A D representing any volume V of free air, and B C the volume V′, to which this quantity of air is compressed; the corresponding absolute pressures being respectively

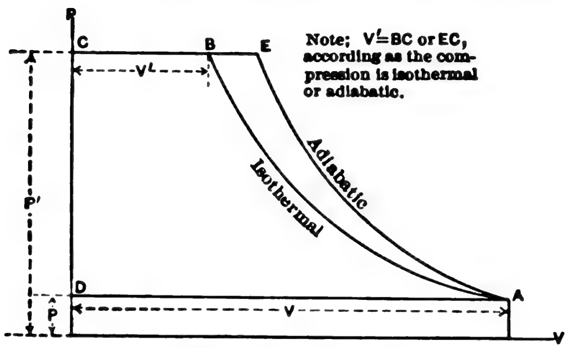
* A statement of the proof of this deduction is unnecessary here; it is given in several books on Thermodynamics, for example, in Perry's work on the Steam Engine, p. 333.

P and P′. The curve is an equilateral hyperbola, and the work, W, represented by the area A B C D = $W_1 + W_2 - W_3$

in which W_1 = area under A B = $\int P\,dV$

W_2 = area under B C = P′ V′, representing the work of expelling the air from the compressing cylinder.

W_3 = area under D A = P V, representing the negative work done by atmospheric pressure on the suction or intake side of the compressing piston.



35.—Reference Diagram.

Since P V = P′ V′, W_2 and W_3 cancel, so that the algebraic sum of

$$W = W_1 + W_2 - W_3 = \int P\,dV \qquad (1)$$

To integrate this expression, substitute for P its equivalent $\frac{P'V'}{V}$ (from the general equation for isothermal compression), whence:

$$W = \int_{V'}^{V} P'V' \frac{dV}{V} = P'V' \int_{V'}^{V} \frac{dV}{V}$$

Integrating: $W = P'V' \times \text{Nap. log.} \left(\frac{V}{V'}\right)$* .... (2)

* The Naperian or hyperbolic logarithm of a number, generally written "log.e," is obtained by multiplying the *common* logarithm by the constant 2.302585.

The equation may also be written:

$$W = P\,V \log_e \left(\frac{P'}{P}\right), \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

which is a form convenient for use in making air compressor calculations. When expressed in foot-pounds (by putting V in terms of cubic feet, and P, P′ in pounds per square inch), the equation takes the form:

$$W = 144\,P\,V \log_e \left(\frac{P'}{P}\right) \quad \ldots\ldots\ldots\ldots \quad (4)$$

If V, the intake capacity of the cylinder in cubic feet, be denoted by L, the equation becomes:

$$W = 144\,P\,L \log_e \left(\frac{P'}{P}\right) \quad \ldots\ldots\ldots\ldots \quad (5)$$

which is the general equation for the work of compressors operating isothermally and without clearance.

Work of Compressors without Clearance. Adiabatic Compression. The expression for the work done in compressing air adiabatically is deduced as follows, referring to Fig. 35. The line A D represents the initial volume, V, of air at normal atmospheric pressure, to be compressed, and the line E C the final volume V′, occupied by the same quantity of air at the end of the stroke of the piston. In undergoing this change of volume, the pressure increases from P to P′ (absolute pressures), and the resulting compression line A E is an adiabatic curve, following the law:

$$P\,V^n = P'\,V'^n = C \text{ (constant), or } P = \frac{C}{V^n} \quad \ldots\ldots \quad (6)$$

In terms of foot-pounds, the total work done in the compressing cylinder is:

$$W = (W_1 + W_2 - W_3) \quad \ldots\ldots\ldots\ldots \quad (7)$$

in which:

W_1 = the work of compression.

W_2 = work required to force the compressed air out of the cylinder, into the receiver.

W_3 = work done by atmospheric pressure on the suction side of the piston, while the inlet air is entering the cylinder.

First.—The work W_1, in foot-pounds, done during compression, is represented by the integral expression:

$$W_1 = \int_{V'}^{V} 144\, P\, d\, V$$

Substituting the value of P, now expressed in pounds per square inch, from equation (6):

$$W_1 = 144 \int_{V'}^{V} \frac{C\, d\, V}{V^n} \qquad (8)$$

Integrating between the limits V and V':

$$W_1 = 144\, C \left[\frac{V^{(1-n)} - V'^{(1-n)}}{1 - n} \right] \qquad (9)$$

Dividing the second member of the equation by (− 1) and substituting for C its value $P\, V^n$:

$$W_1 = \frac{144\, P\, V^n}{n - 1} \left[\frac{1}{V'^{(n-1)}} - \frac{1}{V^{(n-1)}} \right] \qquad (10)$$

$$= \frac{144\, P\, V}{n - 1} \left[\frac{V^{(n-1)}}{V'^{(n-1)}} - 1 \right] \qquad (11)$$

But, since $\frac{P'}{P} = \frac{V^n}{V'^n}$, $\frac{V}{V'} = \left(\frac{P'}{P}\right)^{\frac{1}{n}}$, which, raised to the $n - 1$ power, gives:

$$\frac{V^{(n-1)}}{V'^{(n-1)}} = \left(\frac{P'}{P}\right)^{\frac{n-1}{n}} \qquad (12)$$

Substituting this value in (11):

$$W_1 = \frac{144\, P\, V}{n - 1} \left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1 \right] \qquad (13)$$

Second.—The work W_2, of expelling the air from the cylinder,

$$= 144\, P'\, V' \qquad (13a)$$

Multiplying by $\frac{V'}{V}$ both members of the expression, $\frac{P'}{P} = \left(\frac{V}{V'}\right)^n$:

$$\frac{P'\, V'}{P\, V} = \left(\frac{V}{V'}\right)^{n-1}; \text{ whence, } P'\, V' = P\, V \left(\frac{V}{V'}\right)^{n-1}.$$

But,

$$\frac{V}{V'} = \left(\frac{P'}{P}\right)^{\frac{1}{n}} \text{ and } \left(\frac{V}{V'}\right)^{n-1} = \left(\frac{P'}{P}\right)^{\frac{n-1}{n}}; \text{ hence}$$

$P' V' = PV \left(\frac{P'}{P}\right)^{\frac{n-1}{n}}$; which, substituted in equation (13*a*), gives:

$$W_2 = 144\, P' V' = 144\, P V \left(\frac{P'}{P}\right)^{\frac{n-1}{n}} \quad \ldots\ldots\ldots \quad (14)$$

Third.--The work W_3, done by atmospheric pressure on the back of the piston, $= 144\ P\ V$ .... (15)

Taking the algebraic sum of W_1, W_2 and W_3, from equations (13), (14) and (15), and substituting in equation (7):

$$W = 144 \left\{ \frac{P V}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] + P V \left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - P V \right\};$$

whence, by reducing to a common denominator:

$$W = 144\, \frac{P V \left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] + (n-1)\, P V \left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - (n-1)\, P V}{n - 1}$$

and cancelling:

$$W = \frac{144\, P V n}{n - 1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \quad \ldots\ldots\ldots \quad (16)$$

which is the general expression for the work of single-stage compressors, with adiabatic compression, and when clearance is zero.

The relations between the two conditions of compression are represented graphically by Fig. 36. By laying off to scale the volumes of air on the horizontal line of the diagram, the corresponding pressures at different points of the stroke of the compressing piston are measured on the verticals. The adiabatic curve rises more rapidly than the isothermal, according to the law. Therefore, in compressing adiabatically a quantity of air to a given volume, more work is expended than if the compression were effected isothermally. Perfect isothermal compression cannot be attained in practice. Even with the best cooling arrangements the

compressor would have to run at an extremely slow speed, and be of very large size, to approach closely the condition of isothermal compression. On the other hand, if the air compressed adiabatically could be kept hot until used, the loss of the additional work which was expended in compressing it would be prevented. But neither can this be done. The air is almost always conveyed to considerable distances before it is used,

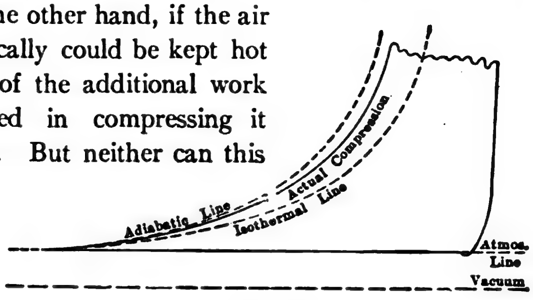


FIG. 36.

and the loss of heat by radiation from the pipes soon reduces the pressure to that corresponding with the temperature of the surrounding atmosphere. In practice, therefore, neither of these theoretical methods of compression is possible; a combination or modification of the two is employed, the net result depending upon the degree of perfection of the compressing engine and of the cooling arrangements provided.

As shown by Fig 36, the actual line of compression must lie somewhere between the adiabatic and isothermal lines provided there is no leakage past the piston. When compressing in a single cylinder to sixty or eighty pounds' pressure, and at a piston speed not exceeding 300 feet per minute, it is probable that about one-half of the total possible cooling is all that may be expected.* The aim is to begin compression with the air at a low initial temperature, and to bring the compression line as close as possible to the isothermal line. Next, it is of the utmost importance that the air shall be cooled thoroughly during compression and before it leaves the cylinder. Any subsequent cooling, whether in the receiver or in the air main, must entail loss.

As a matter of fact, the abstraction of heat during compression in ordinary practice is very imperfect. Some distance must be traversed by the piston, in compressing the air, before there is any considerable rise in temperature, and until the temperature does

* Frank Richards, "Compressed Air," p. 66.

rise no cooling can be effected. In other words, the abstraction of heat does not begin at the beginning of the stroke. The temperatures of the air taken into the cylinder and of the water used for cooling are likely to be nearly the same, so that all the possible reduction of temperature in any one cylinderful of air must take place in a period of time less than that occupied in making the stroke. Most of the cooling is done necessarily in the latter half of the stroke. It should be noted, moreover, that soon after the compressor begins running the cylinder itself becomes quite hot and heats the air during intake. For this reason the total amount of cooling to be effected is greater than that which is required to abstract the heat developed during the compression of a given volume of air to a given tension. In modern dry compressors of fairly large size, and running at full working speed, the compression line is usually much nearer the adiabatic than the isothermal curve, and often follows the adiabatic curve quite closely.

There are two methods of absorbing the heat produced by compression:

1. By introducing cold water into the air cylinder.

2. By cooling the cylinder from without, enveloping it in a cold-water jacket.

Machines of the first class are known as "wet compressors"; those of the second, "dry compressors."

The values of the coefficient n in the equation already given, $\frac{P'}{P} = \left(\frac{V}{V'}\right)^n$, have been found for the different systems of compression. As has been stated, in the case of purely adiabatic compression, with no cooling arrangements, $n = 1.406$; in ordinary single-cylinder dry compressors, provided with a water-jacket, n is roughly 1.3, while in the best single-stage wet compressors (with spray injection) n becomes 1.2 to 1.25. In the poorest forms of compressor the value $n = 1.4$ is closely approached. It should be added that for large well-designed compressors with compound air cylinders and efficient intercooling, the exponent n, referred to the combined indicator cards, may be as small as 1.15. This result has been obtained, for example, from a 2,000 horse-power, two-stage compressor at Quai de la Gare, Paris.

The diagrams, Figs. 37, 38, and 39, show the relative positions of the several compression lines, the areas between the compression and isothermal lines being shaded in each case. These are not actual indicator diagrams. They are intended approximately to represent the relations between the different lines, under the conditions named.

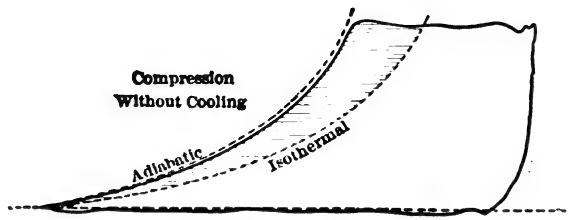


FIG. 37.

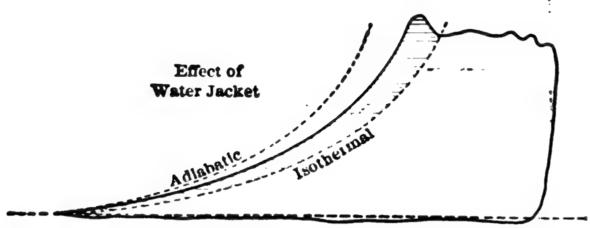


FIG. 38.

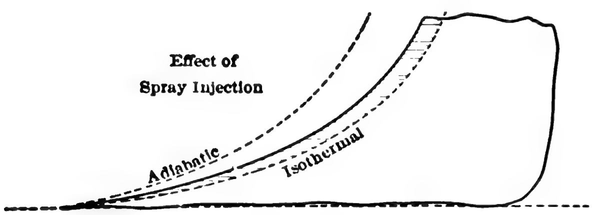


FIG. 39.

Work of Two-stage Compressors, without Clearance.*—In this system of compression the air is brought up to a certain pressure in one cylinder; passes thence to an intercooling chamber, or intermediate receiver, in which the temperature of

* For the construction and operation of stage compressors see Chapter VI.

the air is reduced, and finally enters a second cylinder, where the compression is carried to the desired terminal pressure. It is customary to design the compressor with a cylinder ratio that will divide the work equally between the cylinders, but changes of conditions of operation other than those contemplated may destroy this equality. In the following discussion,* it is assumed that the same quantity of work is done in each stage.

An inspection of the diagram, Fig. 40, shows that there must be some best intermediate receiver pressure, for which the total

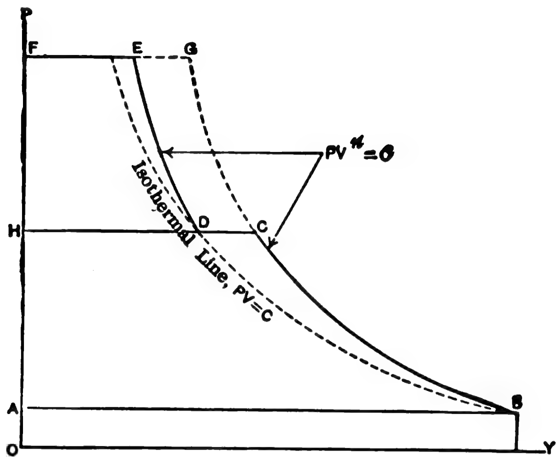


FIG. 40.—Reference Diagram, Two-Stage Compressor, with no Clearance and Perfect Intercooling.

work of compression will be a minimum. For, if this receiver pressure approach either P or P″ (corresponding respectively to the points B and G on the compression curve), then would the compression approach single-stage work and the entire compression line would lie along $\overline{BCG}$. But, with the intercooling receiver at any intermediate point, the broken line $\overline{BCDE}$ is fol-

* I desire here to acknowledge the valuable assistance of Dr. Charles E. Lucke, Professor of Mechanical Engineering in Columbia University, who kindly gave me the use of his original notes on the method of analysis employed in this discussion of the theory of stage compression and the effect of clearance in the different work cycles of adiabatic compression.

lowed, the saving in work over single-stage compression being represented by the area $\overline{\text{C D E G}}$.

The net work of the compressor, W, represented by the area $\overline{\text{A B C D E F}}$, is equal to the work of area $\overline{\text{A B C H}}$ of the first stage plus the work of area $\overline{\text{H D E F}}$ of the second stage, or $W = W_1 + W_2$. Let the condition of perfect intercooling be assumed; that is, the hot air discharged from the first cylinder is cooled in the intermediate receiver to the initial temperature of the intake air. The work cycle in each cylinder is the same as that of single-stage adiabatic compression, as expressed by equation (16), but with two additional symbols for pressures and volumes.

Let A B = V = initial volume of free air in first cylinder.
H D = V″ = initial volume of air in second cylinder.
O A = P = initial absolute pressure (atmospheric pressure).
O H = P′ = terminal absolute pressure in first cylinder, assumed to be also the intermediate receiver pressure and therefore the initial pressure in the second cylinder.
O F = P″ = terminal absolute pressure in second cylinder.

Hence:
$$\left.\begin{aligned} W_1 &= \frac{P V n}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \ldots\ldots \text{first stage} \\ W_2 &= \frac{P' V'' n}{n-1}\left[\left(\frac{P''}{P'}\right)^{\frac{n-1}{n}} - 1\right] \ldots \text{second stage} \end{aligned}\right\} \quad (17)$$

But, assuming the intercooling to be perfect, $P V = P' V''$, whence:

$$W = \frac{P V n}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} + \left(\frac{P''}{P'}\right)^{\frac{n-1}{n}} - 2\right] \ldots\ldots \quad (18)$$

Since the best receiver pressure, P′, is that for which W is a minimum, by differentiating and placing the first differential coefficient $\frac{dW}{dP'} = 0$:

$$\frac{dW}{dP'} = PV\frac{n}{n-1}\left\{\frac{n-1}{n}\,\frac{(P')^{\frac{n-1}{n}-1}}{P^{\frac{n-1}{n}}} - \frac{n-1}{n}\,\frac{(P'')^{\frac{n-1}{n}}}{(P')^{\frac{n-1}{n}+1}}\right\} = 0$$

Whence: $\dfrac{(P')^{-\frac{1}{n}}}{P^{\frac{n-1}{n}}} = \dfrac{(P'')^{\frac{n-1}{n}}}{(P')^{\frac{2n-1}{n}}}$

or $(P')^{-\frac{1}{n}+\frac{2n-1}{n}} = (P\,P'')^{\frac{n-1}{n}}$ from which $P' = \sqrt{P\,P''}$, an expression for the best receiver pressure.

Dividing both terms by P:

$$\frac{P'}{P} = \frac{(P \times P'')^{\frac{1}{2}}}{P} = \left(\frac{P''}{P}\right)^{\frac{1}{2}}$$

But, $\left(\dfrac{P''}{P}\right)^{\frac{1}{2}} = \dfrac{P''}{\sqrt{PP''}} = \dfrac{P''}{P'}$. Substituting these values in equation (17), remembering that $P'\,V'' = P\,V$ and expressing the work in foot-pounds:

$$W = \frac{2 \times 144\,P\,V\,n}{n-1}\left[\left(\frac{P''}{P}\right)^{\frac{n-1}{2n}} - 1\right] \text{........} \quad (19)$$

which is the equation for two-stage compressor work, in terms of the initial volume and initial and terminal pressures, with perfect intercooling and best receiver pressure.

By a method similar to the above, the expression for the work of three-stage compression may also be deduced:

$$W = \frac{3 \times 144\,P\,V\,n}{n-1}\left[\left(\frac{P'''}{P}\right)^{\frac{n-1}{3n}} - 1\right] \text{......} \quad (20)$$

in which P''' is the terminal pressure in the last, or high-pressure, cylinder.

Effect of Clearance in the Compressing Cylinder. In the preceding pages expressions are deduced for the work of compression with no allowance for the clearance volume of the cylinder. From a mechanical engineering and structural point of view, the question of piston clearance is taken up in Chapter VII. It is necessary here to discuss the cycles of operation of single-stage and two-stage compression with clearance. While the work done is the same as that shown by the general equations found for isothermal and adiabatic compression, the work per unit of cylinder volume, or displacement, will be changed, because of the reexpansion of the clearance air. In other words, clearance affects

the volumetric output of the compressor, but not the work of compression per unit of volume of air taken into the cylinder.

Work of Compressors with Clearance. Isothermal Compression. Fig. 41 is a general reference diagram, in which B C represents an isothermal curve.

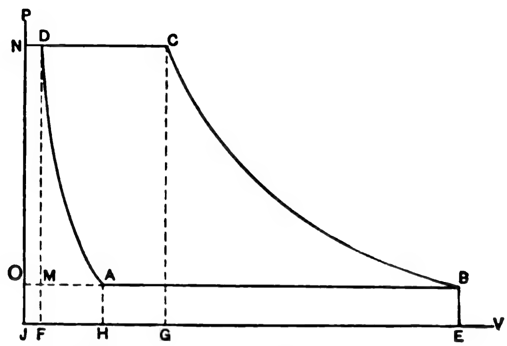


FIG. 41.—Reference Diagram. Compressor Working Isothermally, with Clearance.

Let $E\,B = M\,F = P$
$G\,C = F\,D = P'$
$F\,E = M\,B = V$
$C\,D = V'$
$J\,F = N\,D =$ clearance volume
$FH = (F\,E - H\,E) = M\,A = (M\,B - A\,B) = V'''$, or volume occupied by the re-expanded clearance air.

According to the diagram areas, the total

Net Work $\overline{A\,B\,C\,D}$ = compression work and delivery work $\overline{O\,B\,C\,N}$ − re-expansion work $\overline{O\,A\,D\,N}$.

For the work of a compressor without clearance, and isothermal compression, the general expression, $W = P\,V \log._e \left(\frac{P'}{P}\right)$, has already been deduced. This applies to the areas bounded by the two horizontal lines, the vertical line and the compression line. Similarly, the re-expansion work represented by the area

$\overline{\text{F D A H}}$ (under the curve D A) $= \text{P V}''' \log._e \left(\frac{\text{P}'}{\text{P}}\right)$.

$$\text{Hence, W} = \text{P V} \log._e \left(\frac{\text{P}'}{\text{P}}\right) - \text{P V}''' \log._e \left(\frac{\text{P}'}{\text{P}}\right)$$

$$= \text{P} (\text{V} - \text{V}''') \log._e \left(\frac{\text{P}'}{\text{P}}\right) \ldots\ldots\ldots\ldots\ldots \quad (21)$$

Replacing (V − V‴) by L, which represents the intake capacity of the compressing cylinder, neglecting heating during suction, and expressing P in pounds per square inch:

$$\text{W} = 144 \text{ P L} \log._e \left(\frac{\text{P}'}{\text{P}}\right) \ldots\ldots\ldots\ldots \quad (22)$$

Comparing equations (5) and (22) it is seen that they are identical, as noted above; but it must be remembered that the volume of air actually taken into the cylinder at each stroke and compressed is reduced by the clearance space, and hence the volumetric capacity of the compressor is also reduced. Moreover, neither in this work cycle, nor in that for adiabatic compression, is any account taken of the heating and cooling effects which occur during intake and compression, nor of frictional and other losses which affect capacity and work per unit of air. These points are discussed elsewhere in this chapter and in Chapters IV, V, VI, VII and X.

Single-stage Adiabatic Compression, with Clearance. The reference diagram, Fig. 41, may be used here also, by assuming the line B C to be an adiabatic curve. But, though the work areas are designated as under isothermal compression, and their significations are identical, their numerical values are different.

From equation (16), the work corresponding to area $\overline{\text{O B C N}} =$

$\text{W}_1 = \frac{144 \text{P V} n}{n - 1}\left[\left(\frac{\text{P}'}{\text{P}}\right)^{\frac{n-1}{n}} - 1\right]$; and, similarly, the work corresponding to the area $\overline{\text{O A D N}} =$

$$\text{W}_2 = \frac{144 \text{P V}''' n}{n - 1}\left[\left(\frac{\text{P}'}{\text{P}}\right)^{\frac{n-1}{n}} - 1\right].$$

Whence, $W = \frac{144\,P\,(V - V''')\,n}{n - 1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right]$ .... (23)

Replacing $(V - V''')$ by L (the intake capacity of the cylinder, with clearance):

$$W = \frac{144\,P\,L\,n}{n - 1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \quad \text{.........} \quad (24)$$

Since V in equation (16) may be replaced by L, a comparison of this equation with (24) shows that the work is the same per unit of volume of air admitted to the cylinder; but, the volumetric output is reduced by the clearance.

Though the pressure-volume formulæ serve for most purposes, it is sometimes convenient to have the work expressed in terms of cylinder displacement and volumetric efficiency:

Referring to Fig. 41:

Let D = displacement volume of cylinder in cubic feet, or the effective area × stroke, represented on the diagram by M B = F E.

C = clearance expressed as a fraction of D; whence $C \times D = V_c$ = clearance volume, represented by N D = J F, and $D\,(1 + C)$ = total cylinder volume in cubic feet, represented by J E.

V''' = volume of re-expanded clearance air.

L = intake free air capacity = F E − F H.

E = volumetric efficiency $= \frac{L}{D}$ = the ratio of the length of the actual admission line, A B, to the total distance swept through by the piston.

Then: $V = D\,(1 + C)$, and $V''' = V_c\left(\frac{P'}{P}\right)^{\frac{1}{n}} = C\,D\left(\frac{P'}{P}\right)^{\frac{1}{n}}$

Whence $V - V''' = D\left[1 + C - C\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right] = L$

Substituting this value of $V - V'''$ in equation (23):

$$W = \frac{144\,P\,n}{n - 1}\,D\left[1 + C - C\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right]\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \quad . \quad (25)$$

which expresses the work in terms of displacement and clearance, with pressures in pounds per square inch.

Since $E = \frac{L}{D}$, $L = E\,D = D\left[1 + C - C\left(\frac{P'}{P}\right)^{\frac{1}{n}}\right]$, as above;

whence, substituting in (25):

$$W = \frac{144\,P\,n}{n-1}\,E\,D\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right] \quad \ldots\ldots\ (26)$$

Work of Two-stage Compressor, with Clearance. (Fig. 42.) By a method of deduction similar to the preceding, it may be shown

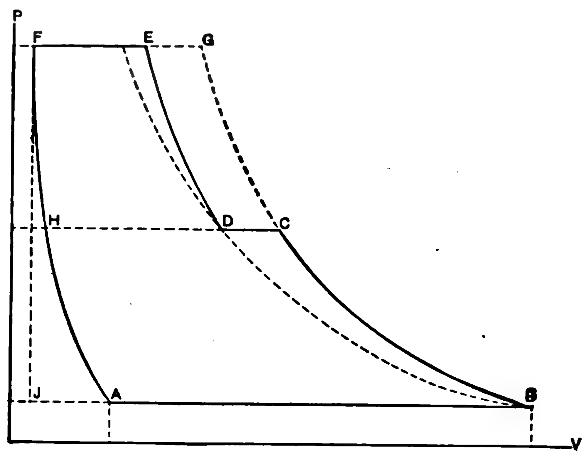


FIG. 42.—Reference Diagram. Work of Two-Stage Compressor, with Clearance.

that the best intermediate receiver pressure for two-stage compression with clearance is the same as that for stage compression without clearance. This follows because the receiver pressure is a function of the compression line and not of the re-expansion line. It will be found, also, that, unlike the work cycle for stage compression without clearance, there is here an unequal division of work between the two cylinders. (See also Chapter VI.)

The general equation for two-stage compression, with propor-

tionate clearance in both cylinders and perfect intercooling at best receiver pressure, is *:

$$W = \frac{144\,P\,n}{n-1}\left\{ 2\,V\left[\left(\frac{P''}{P}\right)^{\frac{n-1}{2n}} - 1\right] - V'''\left[\left(\frac{P''}{P}\right)^{\frac{n-1}{n}} - 1\right]\right\} \quad (27)$$

The diagram (Fig. 42) shows a single, continuous re-expansion line, F A, which is taken to represent the re-expansion lines of both cylinders. This evidently is true only when the clearances of the two cylinders are proportionate. But the cylinders of stage compressors may, and usually do, have different clearances; so that the net high-pressure cylinder volume, at the end of re-expansion of the clearance air is not equal to that of the low-pressure or intake cylinder at the beginning of re-expansion. Nevertheless, since the volume of air delivered by the low-pressure cylinder must necessarily be equal to the volume received by the high-pressure cylinder, disproportionate clearance does not affect the work done per unit of air compressed. The diagram of the high-pressure is merely displaced somewhat with respect to that of the low-pressure cylinder, as shown by Fig. 43, in which F E = F′ E′ and H D = H′ D′.

For disproportionate clearance, therefore, the work is expressed by the equation:

$$W = \frac{144\,P\,n}{n-1}\left\{ (V - V''')\left[\left(\frac{P''}{P}\right)^{\frac{n-1}{2n}} - 1\right] + (V'' - V)\left(\frac{P''}{P}\right)^{\frac{1}{2}}\right\} \quad (28)$$

In equations (27) and (28) the symbols have the following significations:

P and V = initial absolute pressure in pounds per square inch and initial volume in cubic feet.

P″ and V″ = terminal absolute pressure and initial volume of air in the high-pressure cylinder.

V‴ = volume of the re-expanded clearance air (= J A in Fig. 42 and N A in Fig. 43).

* For the sake of brevity, the steps in the deduction of the following work formulæ are omitted. Those readers who desire to pursue the subject further will find a full discussion in Dr. Charles E. Lucke's forthcoming work on *Engineering Thermodynamics*.

The work and capacity of two-stage compressors with clearance may also be expressed in terms of displacement and volumetric efficiency.

Let D_1 and D_2 = cylinder displacements, respectively of the low- and high-pressure cylinders.

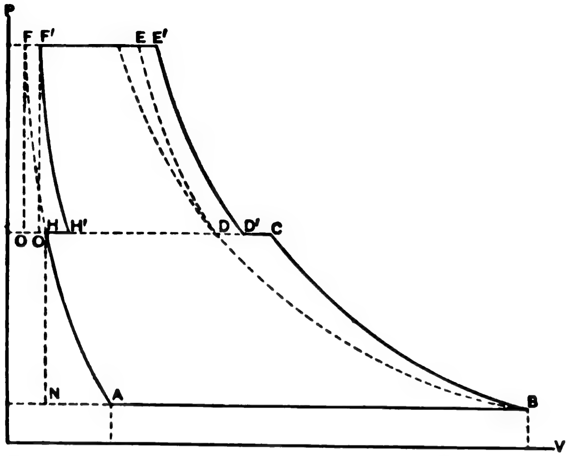


FIG. 43.—Reference Diagram. Work of Two-Stage Compressor, with Disproportionate Clearance.

C_1 and C_2 = fractional clearances; whence $C_1 \times D_1$ and $C_2 \times D_2$ = the clearance volumes, and $D_1\ (1 + C_1)$ and $D_2\ (1 + C_2)$ = the total cylinder volumes in cubic feet.

$$E_1 = \frac{A\,B}{N\,B} = \text{volumetric efficiency of low-pressure cylinder.}$$

$$E_2 = \frac{H'\,D'}{O'\,D'} = \text{volumetric efficiency of high-pressure cylinder.}$$

Introducing these symbols:

$$W = \frac{144\,P\,n}{n-1}\left\{D_1\left[1 + C_1 - C_1\left(\frac{P''}{P}\right)^{\frac{1}{2n}}\right] + \right.$$

$$\left. D_2\left(\frac{P''}{P}\right)^{\frac{1}{2}}\left[1 + C_2 - C_2\left(\frac{P''}{P}\right)^{\frac{1}{2n}}\left(\frac{P''}{P}\right)^{\frac{n-1}{2n}} - 1\right]\right\} \ldots \quad (29)$$

which expresses in foot-pounds the work of the compressor in terms of the displacements and clearances of both cylinders. As the deduction is based on the assumption of best intercooler, or intermediate receiver, pressure, the displacements of the two cylinders must have a corresponding relation, to produce the most economical division of work between them. This relation in turn involves the clearances and the compression ratio.

Referring again to Fig. 43, and denoting pressures and volumes by subscripts identical with the letters on the diagram:

$$D_2 = H' D' + H' O' = H D + H' O'.$$

But $$H D = V_d - V_h = V_b \frac{P_b}{P_c} - V_h = \frac{D_1 (1 + C_1)}{\left(\frac{P''}{P}\right)^{\frac{1}{2}}} - C_1 D_1$$

and $$H' O' = V_{h'} - V_f = V_f \left(\frac{P_f}{P_h}\right)^{\frac{1}{n}} - V_f = C_2 D_2 \left[\left(\frac{P''}{P}\right)^{\frac{1}{2n}} - 1\right]$$

whence: $$D_2 = \frac{D_1 (1 + C_1)}{\left(\frac{P''}{P}\right)^{\frac{1}{2}}} - C_1 D_1 + C_2 D_2 \left[\left(\frac{P''}{P}\right)^{\frac{1}{2n}} - 1\right]$$

or, $$D_2 - C_2 D_2 \left[\left(\frac{P''}{P}\right)^{\frac{1}{2n}} - 1\right] = D_1 \left[\frac{1 + C_1}{\left(\frac{P''}{P}\right)^{\frac{1}{2}}} - C_1\right]$$

Hence, $$\frac{D_1}{D_2} = \left(\frac{P''}{P}\right)^{\frac{1}{2}} \left[\frac{1 + C_2 - C_2 \left(\frac{P''}{P}\right)^{\frac{1}{2n}}}{1 + C_1 - C_1 \left(\frac{P''}{P}\right)^{\frac{1}{2}}}\right] \quad \ldots\ldots\ldots\ldots (30)$$

This equation expresses the ratio between the displacements of the low- and high-pressure cylinders, in terms of the initial and terminal pressures and the clearances.

It may be desirable also to express the work in terms of the volumetric efficiencies. From the diagram:

$$E_1 = \frac{V_b - V_a}{V_b - V_h} = \frac{D_1 \left[1 + C_1 - C_1 \left(\frac{P''}{P}\right)^{\frac{1}{2n}}\right]}{D_1} = \left[1 + C_1 - C_1 \left(\frac{P''}{P}\right)^{\frac{1}{2n}}\right]$$

and similarly: $E_2 = \left[1 + C_2 - C_2 \left(\frac{P''}{P}\right)^{\frac{1}{2n}}\right]$

Substituting E_1 and E_2 for their equivalents in equation (29):

$$W = \frac{144\,P\,n}{n-1}\left\{D_1\,E_1 + D_2\,E_2\left(\frac{P''}{P}\right)^{\frac{1}{2}}\left[\left(\frac{P''}{P}\right)^{\frac{n-1}{2n}} - 1\right]\right\} \quad . \; (31)$$

The capacity, L, of the compressor is measured by the intake capacity of the low-pressure cylinder, or:

$$L = V_b - V_a = D_1\left[1 + C_1 - C_1\left(\frac{P''}{P}\right)^{\frac{1}{2n}}\right] \quad \ldots . \; (32)$$

expressed in terms of displacement and clearance; and since

$$E_1 = \frac{V_b - V_a}{V_b - V_h}, \; L = D_1\,E_1.$$

Finally, if equation (29) be divided by equation (32) the work per unit of intake capacity of the compressor is expressed in terms of cylinder displacements and clearances.

The subject of the performance of air compressors is presented in Chapter X. Tables are there given, showing the work actually required per foot of free air, for single-, two- and three-stage compression, including the work consumed by frictional and other losses.

CHAPTER IV

WET COMPRESSORS

Although during the past fifteen years wet compressors have become almost obsolete in the United States, it is necessary to give some attention to them, not only because many are still used in Europe, but also because a discussion of their design and operation will lead to a better understanding of the comparative merits of the systems of cooling employed in the modern dry compressors.

Wet compressors are of two kinds:

1. The so-called hydraulic-plunger compressors, in which water is introduced in bulk into the air cylinder, and is injected also in the form of spray.

2. Those in which the cooling water is injected in the form of fine spray or jets only.

Compressors of the first type comprise some of the earliest forms of air compressor. One of the best of this class is the modernized Dubois-François, built at Seraing, Belgium. It has been widely used in Europe, for mining and tunnelling operations, and it is worth noting that, up to about 1877, one of them was employed at the Sutro tunnel, Nevada. Another well-known compressor of the same class, but of different design, is the Humboldt, made at Kalk, near Cologne, Germany. One of these also was erected at the Sutro tunnel, and did excellent work. A brief description of the old Humboldt compressor (Fig. 44) will serve to explain the principle and construction of these machines.

The water constitutes a piston for compressing the air; an ordinary plunger, like that of a pump, moving in a horizontal cylinder filled with water. At each end of the cylinder, and connected with it by an easy curve, is a vertical air chamber. The upper ends of these chambers are provided with the necessary air

inlet and discharge valves. As the piston reciprocates, the air is drawn alternately into one air chamber and compressed in the other, by the rise and fall of the water level. At the end of each stroke the air compressed by the rising mass of water in the air chamber passes through the discharge valves into the receiver. As the air is in contact with the water a partial cooling is effected, and to prevent the water itself from becoming heated a constant circulation must be maintained. A further cooling is brought about by the injection of sprays from a small force pump into the cylinder and vertical air chambers. The pump is operated from

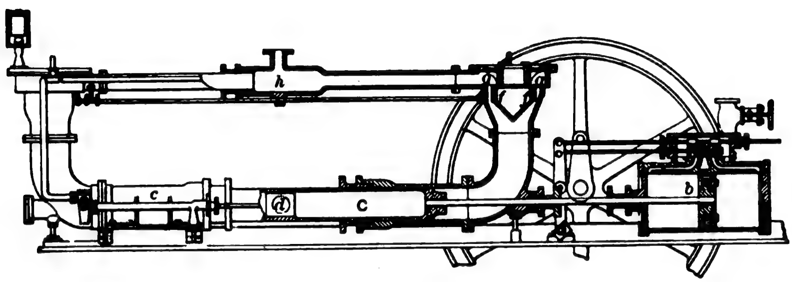


FIG. 44.—Humboldt Wet Compressor.

the cross-head of the compressor itself. This type of compressor is simple, and if the sprays be copious the air is quite effectually cooled; but it is generally limited to rather slow speeds (only 100 to 150 feet piston speed per minute or less in some cases), on account of the inertia of the body of water. This is about one-third to two-fifths of the piston speed of modern dry compressors, and it follows that such engines are comparatively heavy and bulky for a given output of air, besides requiring expensive foundations. It is claimed, however, that a more recent form of Humboldt wet compressor can be run successfully at speeds of 300 to 360 feet per minute, the temperature of the air at discharge being kept at 77° to 80° Fah.* This is such remarkably good work that the results are open to question, as far as regular, normal service is concerned. Lower speeds are certainly advisable for this form of compressor.

* P. R. Björling, *Colliery Guardian*, Oct. 2d, 1896, pp. 629–630.

The machines are made of large size and are heavily and substantially built. Violent shocks are apt to be caused by attempting to run at high speeds, for which reason the vertical air chambers join the cylinder with a curve of long radius to ease the movements of the mass of water.

Fig. 45 shows a late type of the Hanarte wet compressor, many of which have been built for French and Belgian mines, and also for use in connection with ice-making plants. They are generally of large size, and are found to be highly efficient when run at piston

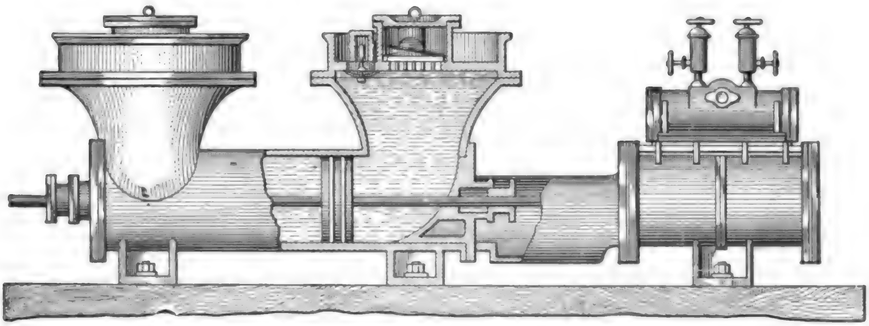


FIG. 45.—Hanarte Wet Compressor.

speeds of 250 to 275 feet per minute. The splayed out vertical ends of the cylinders cause the level of the water to rise slowly towards the end of the stroke, and afford space in the cylinder heads for large and readily accessible inlet and delivery valves. Sprays are used in addition to the water in bulk.

A difficulty with wet compressors of this class is that an efficient circulation of cold water is not easy to maintain. Only a small quantity of fresh water can be injected at each stroke, and without copious sprays the cooling is imperfect. This is due to the fact that, although the mass of water kept in motion in the cylinder and air chambers is large, there is between it and the air only a surface contact. Since water is a poor conductor of heat, under these conditions it can hardly be questioned that the air is cooled more by contact with the relatively large area of the wet cylinder walls than by its contact with the small superficial area of the rising and

falling water. Another disadvantage is that the compressed air delivered from the cylinder is practically saturated with moisture.

Compressors of the second class, in which the cooling water is used only in the form of jets or spray, constitute an improvement upon the older design, in being much less cumbrous and permitting a higher working speed. This method of cooling was first applied by Colladon, at the St. Gothard tunnel. Though these compressors are still frequently used in Europe, they have given way in great measure to dry compressors, and in American practice have become almost obsolete. The air cylinder does not differ materially from that of the dry compressor. A small water pipe is tapped into each cylinder head and fine spray is injected in front of the piston while compression is taking place.

Undoubtedly this system is superior to that involving the use of water in bulk. Since the water is in a state of fine division a relatively large surface of contact is presented, and the air is kept thoroughly saturated with moisture during compression. Zahner, in his "Transmission of Power by Compressed Air," p. 28, states that Colladon's St. Gothard compressors, "which were run at a piston speed of 345 feet, and compressed the air to an absolute tension of 8 atmospheres (103 pounds gauge pressure), gave an efficiency which never descended below 80 per cent, while the temperature of the air never rose higher than from 12° to 15° C. (53° to 59° F.)." The temperature of the injection water is not stated, but must have been very low to obtain such remarkable results.

A dry compressor may be converted into a wet compressor merely by providing the cylinder with water jets. The injected water collects in the cylinder until enough is present to fill the piston clearance space at the end of the stroke. Then any additional amount of water is forced out at each stroke with the compressed air through the discharge valves into the air receiver. From the receiver the water is drained away from time to time. As the piston clearance in well-designed compressors is extremely small, very little water can remain in the cylinder to be churned back and forth by the piston. The water used for injection should be as pure and

cold as possible. Gritty water must never be employed, as it would injure the cylinder, piston and valves.

A proper injection apparatus should fulfil three conditions:

1. The injection must commence at the beginning of the stroke and continue to the end, against the advancing piston.

2. There should be a thorough diffusion of the water in the form of spray throughout the cylinder. By mere surface contact water takes up but little heat. Even a single strong jet is quite effectual, however, because on striking the piston it is thoroughly broken into spray.

3. A definite volume of water should be injected, the quantity increasing with the pressure under which the compressor is working; that is, with the quantity of heat generated. If the quantity of water used be insufficient to abstract the heat, a large amount of moisture is taken up by the warm air and carried into the receiver and piping.

The heat units developed by compression having been calculated, the quantities of water required for different pressures are shown in the following table.* The average temperature of injection water may be taken as, say, 68° Fah., and is considered

TABLE III

Pressures.		Heat units developed by compression in one pound of free air.	Pounds of water to be injected at 68° F. to keep final temperature at 104° F.	
Above vacuum. Atmospheres.	Gauge pressure. Pounds.		Per pound of free air.	Per cubic foot of free air.
2	14.7	58.310	0.734	0.056
3	29.4	92.390	1.164	0.089
4	44.1	116.627	1.469	0.112
5	58.8	135.388	1.701	0.130
6	73.5	151.700	1.891	0.144
7	88.2	163.735	2.063	0.158
8	102.9	174.937	2.204	0.168
9	117.6	184.865	2.329	0.178
10	132.3	193.701	2.440	0.186
12	161.7	209.090	2.634	0.201

* This table is taken in part from that given by Zahner, "Transmission of Power by Compressed Air," p. 110, English units being substituted for French.

as having accomplished its work if it leaves the cylinder at 104° Fah., these temperatures corresponding, respectively, to 20° and 40° C.

There is no practical advantage to be gained by using an excessive quantity of water, and care should be taken to inject no more than is required. The additional cooling effect of a greater mass of water in the cylinder would be but small—as has been remarked under wet compressors of the first type—and more power would be consumed in pumping the water into the cylinder and then forcing it out again through the delivery valves.

CHAPTER V

DRY COMPRESSORS

In the dry system of compression no water enters the air cylinder except that which is carried as moisture in the air itself. All the cooling during compression, aside from radiation, is effected by a water envelope, or "jacket," surrounding the cylinder, and in which cold water is kept constantly circulating.

Fig. 46 shows the longitudinal section of a Nordberg jacketed air cylinder. (Reference may also be made to Figs. 2, 5, 8, 11, 23, and other cuts of longitudinal sections, as illustrating different types of jacketed cylinders.) The cylinder is enclosed in an outer shell, leaving an annular space, J J, to be occupied by the water. Besides the annular jacket nearly one-half the area of each cylinder head is also covered by water jackets, K K. The remainder of the end areas is occupied by the suction and delivery valves, as shown. The air-delivery valves are sometimes placed radially, close to the cylinder ends, whereby a larger proportion of the area of the heads can be jacketed. This is true, for example, of one or two of the Laidlaw-Dunn-Gordon patterns.

In Fig. 46 the circulation of water is effected by pipes connecting with the openings A and B, respectively for inlet and discharge. To cause a proper circulation the spaces enclosed by the jacket are subdivided. The cold water enters at A, and after circulating through the annular and end jackets J J, K K, is finally discharged at B. The smaller jackets on the cylinder heads are designed to surround the valves and air passages as completely as possible, in order to exert the maximum degree of cooling. At C is a drain pipe through which the jacket is blown out occasionally to clear it of sediment.

In some makes of compressor, the annular jacket is divided by vertical partitions, so that the cold water entering at the top passes first around about one-fifth of the length of the cylinder nearest each end. The water then circulates around the middle portion, and is discharged at the top. Although in this arrangement the fact is recognized that at the end of the stroke, where the air pressure is highest, the greatest amount of heat is generated; still, in some of the same designs little, if any, of the cylinder-head

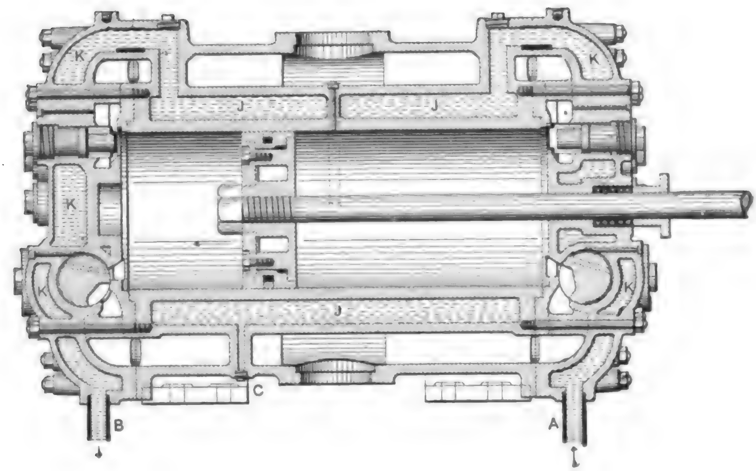


FIG. 46—Air Cylinder of Nordberg Compressor.

area is jacketed, because of the mode of placing the inlet and discharge valves. This would seem to be a defect because, on approaching the end of the stroke, the piston rapidly covers the annular jacket, leaving a very small part of its area available for cooling the hot compressed air while being discharged from the cylinder. It is at this point of the stroke that large end jackets are most valuable. Similar provision for large jacket surfaces is made in the Ingersoll-Rand compressors, particularly those of the "Hurricane Inlet" type. The water passes first into the jackets on the cylinder heads and then successively through several separate compartments of the annular jacket.

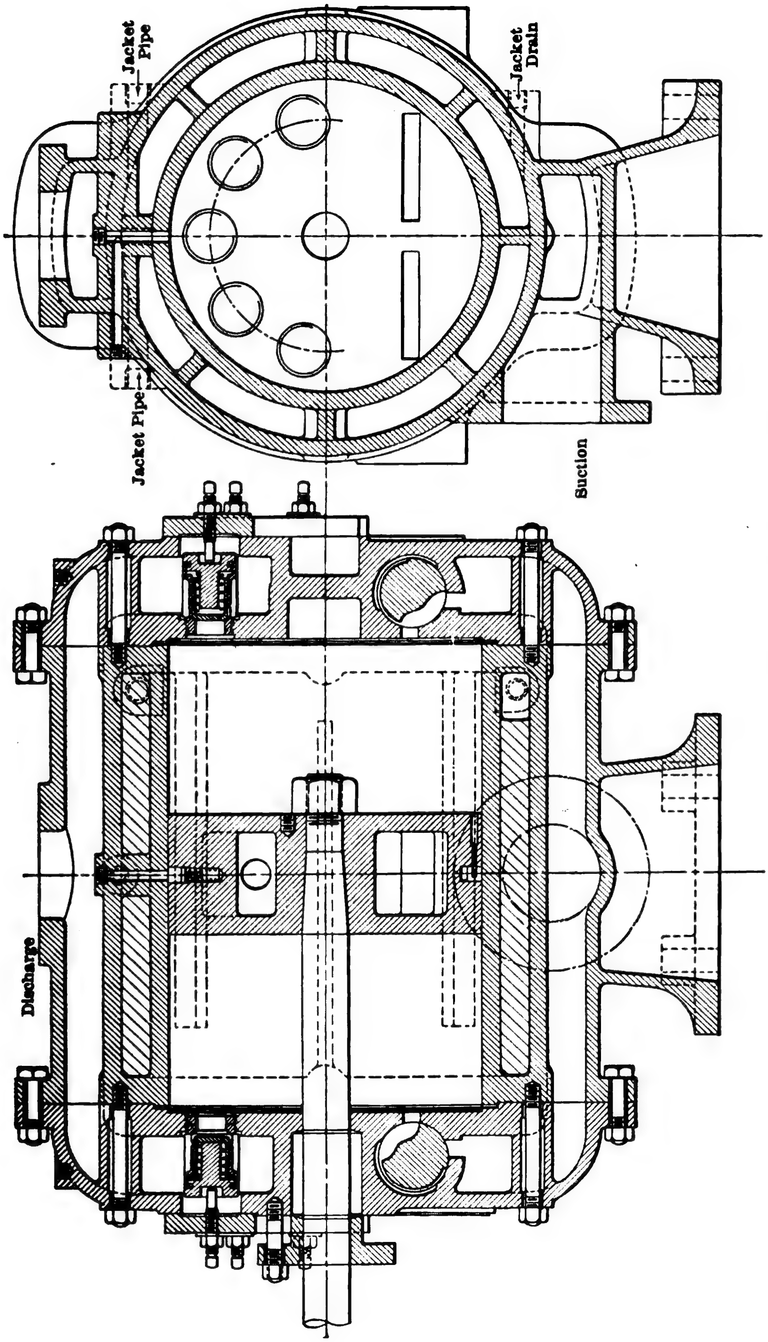


FIG. 47.—Air Cylinder, Class E, Laidlaw-Dunn-Gordon Co.

The jacket of one of the Laidlaw-Dunn-Gordon designs (Fig. 47) is cast with eight longitudinal partitions, extending alternately from each end of the cylinder nearly to the opposite end. The water, which enters near the top, is forced to travel back and forth between the partitions and from one end of the cylinder to the other until it is finally discharged. An active circulation is thus maintained. For furnishing the cooling water a tank is often provided, set at some elevation above the compressor, or a small pump may be employed.

Naturally, a partial cooling only can be effected by water-jacketing the air cylinder. Much depends on the speed at which the compressor is run. In the best single-stage compression, to say seventy or seventy-five pounds, and at not over 300 feet piston speed, it is doubtful whether more than about one-half of the total possible cooling can be effected; that is, in the equation $\frac{P'}{P} = \left(\frac{V}{V'}\right)^n$, n would be equal to, say, 1.22 to 1.25. Heat is generated faster than it can be abstracted, and only a portion of the volume of air passing through the cylinder comes into direct contact with the cooling surfaces. It is important, therefore, that as much as possible of the total cylinder surface be covered by the jacket, and that the piston speed be moderate. But, in a dry compressor, as the air is comparatively free from moisture, some heating is not so objectionable as it would be in a wet compressor. As a matter of fact, the cylinder, discharge pipe, and even the receiver, are usually quite hot when the compressor is running at full speed; often too hot to be touched with the hand. In a plant at Birmingham, England, with well-jacketed cylinders, and compressing only to forty-five pounds, a temperature of the air at delivery has been observed as high as 280° F. In this case the compressor is large, so that the superficial area of the jackets is small as compared with the volume of the cylinder. It is probable that the heat of compression in dry compressors ranges from 200° to a maximum of 400° F. for the ordinary pressures used in mining, though it does not often exceed 350°. Care should be taken not to allow the temperature

to rise above this point.* At a large mine in Montana, the writer has observed the thin wrought-iron delivery pipe of a fifty-drill compressor red-hot for a distance of nearly six inches from the cylinder shell. Driving compressors at too high a speed (when not large enough for their work) is often the cause of the poor results complained of by some users of compressed air.

In some compressors the inner shell of the air cylinder, *i.e.*, between the cylinder and water-jacket, has been made of hard brass, which by its high conductivity assists in carrying off the heat. With the same end in view, the cylinder walls should be as thin as is consistent with safety.

Besides its function of cooling the air during compression, the water-jacket of a dry compressor is indispensable from a mechanical point of view, in keeping down the temperature of the cylinder shell. Without some special provision for cooling the cylinder the metal would become hot enough to burn the oil, and render proper lubrication impossible. To furnish a larger cooling surface one of the older styles of Rand compressor had a hollow back piston-rod and hollow piston, through which water is circulated. To maintain circulation the back piston-rod worked telescopically in a stationary tube connected with the water supply.

Piston Clearance in the Air Cylinder. In every engine, whether steam engine or compressor, the amount of clearance at the end of the stroke, between the piston and cylinder head, is a matter of some importance. It has a special bearing in the case of a dry compressor, which may be explained as follows. Toward the end of the stroke the compressed air in front of the piston begins to pass through the delivery valves as soon as its tension exceeds that of the air in the discharge pipe leading from the cylinder to the receiver. But remaining in the clearance space, on the completion of the stroke, is a certain quantity of warm compressed air, which in the case of a dry compressor can never be discharged. On the back stroke the clearance air expands and partly fills the cylinder behind the piston. No air can enter through the inlet valves until the pressure inside the cylinder falls below atmos-

* T. G. Lees, *Trans. Federated Inst. Mining Engrs.*, Vol. XIV, p. 569. See also Chapter XIII of present volume.

pheric pressure. It is never possible, therefore, to take a full cylinder of fresh air even under the best conditions, and the clearance space must be made as small as possible, say, about one-sixteenth inch. Or, the clearance may be expressed as a ratio, by dividing the clearance volume by the entire cylinder volume swept through by the piston in making its stroke. In a wet compressor the clearance space is filled with water, and therefore does not produce the effect just described.

In cylinders of the same diameter and having the same amount of linear clearance at the ends of the stroke, it is evident that the ratio between cylinder volume and clearance volume depends on the length of stroke. This ratio is generally largest in short-stroke compressors and smallest in those of long stroke. It varies, also, in compressors of different makers. As examples may be cited the following ratios between cylinder and clearance volumes of several Ingersoll-Rand compressors, of different strokes:

14 inch stroke	.0190
21 " "	.0176
24 " "	.0126
36 " "	.0112
48 " "	.0093

ranging thus from about 2 per cent. down to 1 per cent. Some recent compressors, built by the same company, of 42-inch stroke, but of relatively small cylinder diameters, have piston clearances as small as .78, .80 and .90 of one per cent., and several of 36-inch stroke have clearances of .83 and .84 of one per cent. In a recent type of the Leyner compressor, with a 22-inch cylinder, this percentage is stated to be 1.02, and in several of the Laidlaw-Dunn-Gordon compressors of standard type, it ranges from .75 to 1.25 per cent. Clearances are generally larger, however; thus, in a new design of direct, electric-driven, two-stage compressors, of the Ingersoll-Rand Co., the following clearances are found in the low-pressure cylinders:

28 inch	×	24 inch	....	2.15	per cent.
23 "	×	20 inch	....	1.70	" "
19 "	×	16 "	....	1.85	" "
18 "	×	14 "	....	2.00	" "
17 "	×	14 "	....	2.19	" "

The compressors of some other makes have clearances as high as 2½ per cent. of the cylinder volume.

With few exceptions the lowest figures apply to large, long-

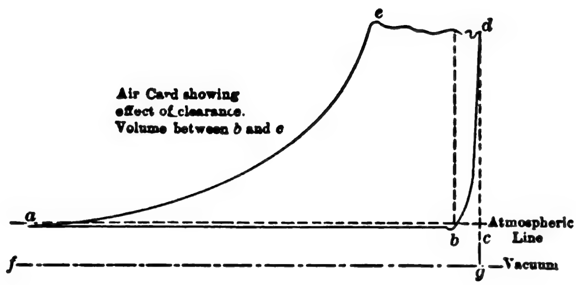


FIG. 48.

stroke compressors; the higher to the small, short-stroke machines in common use for many kinds of service.

The diagram, Fig. 48, shows the effect of clearance. Before the inlet valves can open, the piston must travel from *c* to *b*, and the corresponding cylinder volume passed through by the piston represents the percentage of loss of volumetric capacity as stated above. The actual effect of piston clearance on the volumetric efficiency of the compressor of course depends on the number of compressions; that is, on the air pressure produced. The higher the terminal pressure, the farther must the piston travel before the inlet valves can open and the greater is the distance from *c* to *b*, in the diagram. It may be added, however, that this reduction of capacity, although a matter of considerable importance in the operation of the compressor, does not involve a corresponding loss of useful work. The compressed air remaining in the clearance space helps to

overcome the inertia of the moving parts at the beginning of the return stroke, and to compress the air on the other side of the

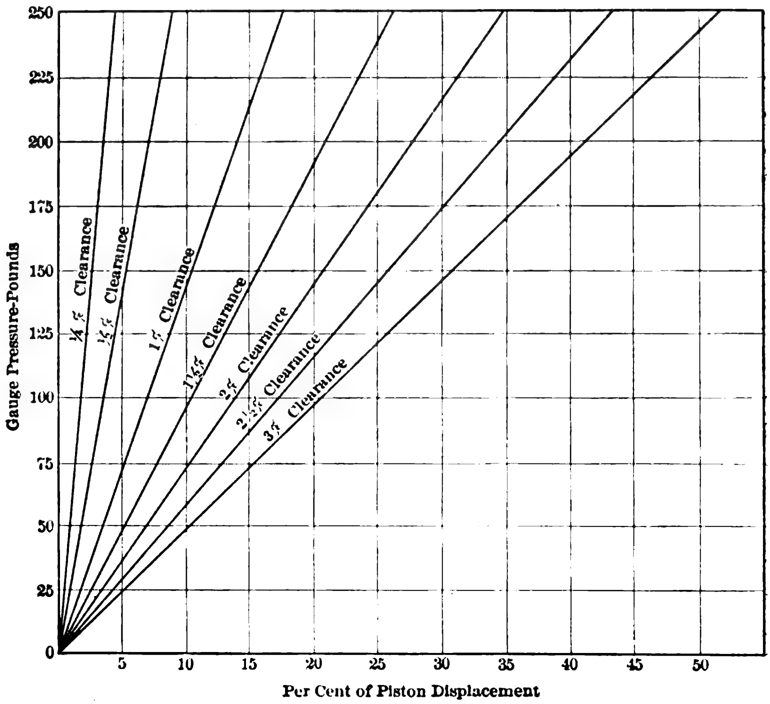


FIG. 49.

piston. A part of the work expended in compressing the clearance air is thus recovered. It has been observed that the clearance air cools slightly during the momentary stoppage of the piston as the stroke is reversed, but the consequent reduction of pressure is a negligible quantity. In expanding behind the retreating piston, however, the clearance air rapidly gives up its heat and does not, therefore, tend to raise the temperature of the incoming atmospheric air.

The effect of piston clearance in reducing the capacity of a dry compressor is shown clearly by the diagram, Fig. 49, which is

reproduced here by kind permission from *Engineering News*, May 30th, 1901. It shows that, for clearances above one per cent. the loss becomes serious even at pressures of seventy-five to one hundred pounds.

Fig. 50 indicates the method of reducing the clearance for ordinary pistons, by casting a recess in the cylinder head to receive the projecting piston nut at the end of the stroke. The loss of volumetric capacity due to clearance of course increases with the air pressure, and in some compressors the piston is run exceed-

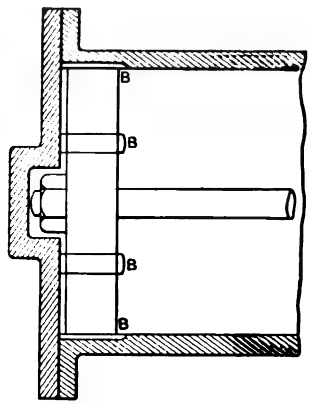


FIG. 50.

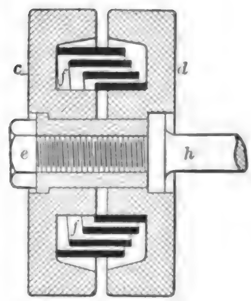


FIG. 51.

ingly close to the cylinder head. When this is the case the compressor must have careful attention, so that if the working length of the connecting rod should be varied in fitting new brasses, the piston will not be in danger of striking the cylinder head.

The Johnson compressor, made in England, has an ingeniously designed piston (Fig. 51) to meet the difficulty just mentioned. It is composed of two disks, *c* and *d*, mounted on a brass sleeve, screwed on the piston-rod, *h*, and held in place by collar and locknut. The disks are so cast as to leave between them a recess, in which is placed a heavy helical spring, *f*. This spring is compressed sufficiently between the disks to prevent it from being further compressed under the maximum working air pressure, but the

clearance at the ends of the stroke is extremely small, and should the piston strike the cylinder head the spring gives slightly and an injurious shock is avoided.*

A number of other devices have been adopted for overcoming the disadvantages of piston clearance. Two examples may be given:

1. Longitudinal bye-pass grooves (B B) are cast in the inner surface of the cylinder near the ends, Fig. 50, so that when the piston reaches the end of its stroke a part of these grooves is uncovered, and the compressed air in the clearance space passes to the other side of the piston.

2. In slide-valve compressors the valve may be provided with a so-called "trick-passage." At the end of the stroke this passage is brought into connection with two small ports entering the extreme ends of the cylinder. Through these passages the high-pressure air in the clearance space is released into the other end of the cylinder.

Although by these methods the released air becomes of direct benefit, there is a decided objection to their employment if all the confined air be allowed to pass over, because the heavy pressure on the piston is suddenly removed, and there is a shock to the moving parts which is clearly evidenced by pounding at the end of the stroke. In the most recent forms of compressor made in the United States the clearance space is very small, but the air confined in it is not released.

Dry Versus Wet Compression

Up to about 1885 there seemed to be little doubt among mechanical engineers that the wet compressors are, upon the whole, superior to the dry, because by bringing the air into direct contact with water the heat is most effectually absorbed. This view is correct so far as heat loss alone is concerned, provided the water introduced into the cylinder is properly applied, as pointed out in

* Björling, *Colliery Guardian*, Aug. 7th, 1896, p. 272.

Chapter IV. Without cooling the percentage of work converted into heat during compression, and therefore lost, is as follows:

Compression to	2 atmospheres,	9.2	%	loss.
" "	3 "	15.0	%	"
" "	4 "	19.6	%	"
" "	5 "	21.3	%	"
" "	6 "	24.0	%	"
" "	7 "	26.0	%	"
" "	8 "	27.4	%	"

In well-designed dry compressors, working at a pressure of 5 atmospheres, the heat loss is reduced about one-half, or from 21.3 per cent. to 11 per cent. Frequently, however, in ordinary mining practice, with single-stage compressors, the loss is fully 15 per cent. By spray injection this loss has been cut down in the best American practice to as little as 3.6 per cent.,* and in some of the large, slow-running European wet compressors to 1.6 per cent. But the question of heat loss is not the only consideration. Low first cost and simplicity of construction are often more advantageous than a close approximation to isothermal compression. Latterly the wet systems have lost ground, and it is probable that no wet compressors are now being built in the United States. In Europe also dry compression has grown in favor, at least for mining plants and others of moderate size. The matter may be considered from two standpoints, as regards:

1. The effect of injected water upon the compressed air and the machines using it.

2. The effect of the water upon the working of the compressor.

In addition, it is necessary to take account of the relative efficiencies of the two types, but this will be deferred until later.

First, it is unquestionable that by using large slow-speed engines, and an abundance of injection water, the air is well cooled, though at a higher first cost for plant. Wet compression gives a good indicator card. It is shown by Table IV that in compressing moist air somewhat less work is expended than for dry air. This

* As stated regarding the old Ingersoll injection compressor, by W. L. Saunders, "Compressed Air Production," p. 24.

is due to the fact that the specific heat of watery vapor is about twice that of dry air; therefore in the presence of moisture more heat is required to raise the temperature of the air in the compressing cylinder, and the loss of work from this cause is reduced.

TABLE IV

Absolute Pressure. Atmospheres.	Gauge Pressure. Pounds.	Foot Pounds of Work Required to Compress One Pound of Air.	
		Dry Compression.	With Sufficient Moisture.
1	0		
2	14.7	23,500	22,500
3	29.4	37,000	35,000
4	44.1	48,500	45,000
5	58.8	58,500	52,500
6	73.5	67,000	60,000
7	88.2	75,000	66,000

Theoretically, a corresponding economy takes place also when the air is expanded again in the machine using it.

Notwithstanding these advantages, several serious objections became apparent in the use of the wet system of compression. Other things being equal, the amount of heat given up during compression is proportional to the difference of temperature between the air taken into the cylinder and the injected water, and to the time of contact between the air and water. Under ordinary circumstances this difference of temperature is zero at the beginning of the stroke, reaching its maximum at the end. It follows: (1) that to attain a fair approach to isothermal compression the piston speed must be very slow; (2) that during the first part of the stroke but little heat is removed, and it is only when compression is complete, and the air begins to pass from the cylinder through the discharge valves, that the cooling effect is at its maximum. At ordinary piston speeds, therefore, a large proportion of the total heat must be given up after the discharge valves have opened; in other words, after compression is completed. For this reason it would appear that, so far as economy of work is concerned, the lower final temperature due to spray injection is in a measure de-

ceptive. The warmth of the air at discharge augments its moisture-carrying capacity, and though it is intended that the separation of the water shall be as complete as possible in the air receiver, still it must of necessity be imperfect in a receiver of any reasonable size. Much moisture passes into the air mains, and deposits as the air cools down in long lines of piping. In cold weather it may freeze so as to reduce the effective diameter of the pipe. The moisture remaining in the air has a further ill effect when it is used. At the instant of exhaust by the drill, or other air engine, the intense cold produced by expansion causes the formation of troublesome accumulations of ice in the exhaust passages.

As to the dry compressor it must be admitted that as air is a poor conductor of heat it has little opportunity to give up its heat of compression between the strokes of the piston. Besides this, the piston, as it advances, rapidly covers the jacket-cooled surface of the cylinder. However, although atmospheric air as taken into the compressor always contains moisture, which will make its appearance as frost at the exhaust of the air machine, still there is not enough of it to cause serious trouble.* The delivery of warm air by a dry compressor is far less objectionable than warm air from a wet compressor.

Second, as to the effect of injected water upon the working of the compressor. Under the best of circumstances water in the air cylinder is objectionable, because it makes lubrication difficult, causes rust, and increasing the wear of piston and cylinder involves greater expense for repairs and renewal of parts. No satisfactory method has ever been devised for lubricating the inner surface of wet compressor cylinders. This is one of the chief difficulties with wet compressors, and becomes most serious when the water is impure or gritty. It must, of course, contain no trace of acid, such as is often present in mine water. Water that is com-

* The quantity of moisture in the atmosphere, or its humidity, varies with the climate, the season of the year, and in a measure with the altitude above sea-level. It is usually greatest near the ocean or any large body of water. What is commonly called dry atmospheric air contains from forty to fifty per cent. of the quantity necessary to saturate it. The degree of saturation in summer often reaches ninety per cent. or more.

paratively harmless for use in jackets might be decidedly injurious to the finished surfaces of working parts. It has been stated by Mr. W. L. Saunders that, although the thermal loss is higher in dry than wet compressors, the frictional loss in the moving parts is considerably higher in the wet compressor. The net economy of the best wet compressors is probably no greater than that of the best American dry compressors.

It is urged on behalf of wet compression that the piston-clearance space is filled with water, and the capacity of the compressor is therefore increased. While this is true, yet, as water is incompressible, and as a part of it must be forced out through the discharge valves at each stroke, the wet compressor is compelled to work in a measure like a water pump. Furthermore, closer attendance is required to regulate the water supply. The drip cock at the bottom of the receiver must also be watched more closely to prevent flooding, and there is the disadvantage of having an injection pump to care for and regulate.

CHAPTER VI

COMPOUND OR STAGE COMPRESSORS

COMPOUND or stage compressors have two or more air cylinders, between which the total work of compression is divided. The air cylinders are placed tandem on a common piston-rod, as in straight-line machines, or respectively tandem with the steam cylinders in the duplex type. In two-stage compressors air at atmospheric pressure is taken into the large or low-pressure cylinder; is there compressed to a certain point, and is then forced into the smaller or high-pressure cylinder, where it is brought up to the required tension (see Figs. 8 to 23). Manifestly, the size of the low-pressure or intake cylinder determines the capacity of the compressor. In a certain sense, the operation of a two-stage compressor is the reverse of that of a compound steam engine.

The theory and application of stage compression are readily comprehended. Since the heat of compression increases with the air pressure produced—though not proportionately, as has been shown—it follows that the higher the pressure the more difficult does it become to keep down the temperature to a point permitting efficient operation of the compressor and proper lubrication of the air cylinder. In attempting, with a single-cylinder dry compressor, to compress even to 90 pounds gauge, the theoretical final cylinder temperature becomes 459° F., and at 100 pounds gauge 485° F. Though some heat is dissipated by radiation, the actual working temperatures corresponding to these pressures are still too high to be dealt with effectually by the ordinary water-jacket, because in a single cylinder the superficial area to which cooling can be applied is too small relatively to the volume of air, and the total compression period too short. Even when working at moderate piston speeds, say, not over 350 to 400 feet per minute, the cooling is very

imperfect. The compressed air, as discharged from the cylinder, is still hot, so that considerable loss of pressure and of work, due to subsequent cooling, is inevitable.

These disadvantages are in large measure overcome by the adoption of stage compression, and, in view of the fact that this system was introduced over twenty-five years ago, it would appear strange that until quite recently it has been neglected, by nearly all compressor builders, for the ordinary pressures used in mining, tunnelling, and similar work.

Formerly it was customary to employ stage compression only when high pressures were required, such as for pneumatic locomotives, riveting machines, presses, compression of gases, pneumatic guns, etc. For such service stage compression is indispensable; and the higher the pressure the greater becomes the necessity for compounding the air cylinders and the comparative efficiency of the system. To produce very high pressures, of 500 to 1,000 pounds or more, three- and four-stage compression is employed.

But it is now generally recognized that two-stage compression when properly applied presents some advantages even for pressures of seventy to eighty pounds, as commonly adopted for machine drills and ordinary air engines. The cooling during compression is more thorough because the total heat generated is divided between two or more cylinders. In each cylinder the temperature is lower than when the same total pressure is produced in a single cylinder, and the combined water-jackets afford a much larger cooling surface.

A further cooling is effected by an "intercooler," placed between the cylinders. This constitutes one of the most important features of stage compression. It is an intermediate cooling-chamber, through which the partially compressed air from the intake or low-pressure cylinder passes on its way to the high-pressure cylinder. The temperature of the air is here reduced, so that when the high-pressure piston begins its work the temperature of the volume of air on which it acts is considerably below that at which the air was discharged from the low-pressure cylinder. Obviously, the

total reduction of temperature effected depends on the volume of the air under compression, the area of the cooling surfaces and the length of time the air is in contact with these surfaces; or, in other words, on the piston speed. The construction of the intercooler will be taken up later.

It should not be inferred from what precedes that stage compression *per se* is always applicable, nor that it is necessarily more economical than compression in a single cylinder. Concerning this, several fairly well defined, though interrelated statements may here be made:

1. Although stage compression is theoretically advantageous for all pressures, it becomes of doubtful utility for gauge pressures of much less than seventy-five pounds, because of the small saving as compared with the greater first cost and running expenses of the more complicated mechanism. It is generally applicable for pressures higher than seventy to seventy-five pounds.

2. Stage compression is specially useful for large compressors, in which the percentage of saving will represent an amount sufficient to warrant the greater first cost of plant.

3. The higher thermodynamic efficiency of stage compression is in some degree offset, and in poorly designed plants may be entirely neutralized, by the increased frictional losses involved in the use of several cylinders. In other words, when employing stage compression, advantage should always be taken of the opportunity to use a well-designed, economically working steam end, together with large and efficient cooling arrangements for the air end. If these requirements be not fulfilled, stage compression may easily cost more per cubic foot of air delivered than simple compression by a properly designed compressor.

Almost all stage compressors are double-acting; that is, on each forward and back stroke air is taken into the cylinders on one side of the piston, while compression and delivery are going on on the other side. The operation of the single-acting form, occasionally employed, will be considered first. It is materially different from that of the double-acting compressor, but its description will aid in setting forth the subject of stage compression.

Single-Acting Two-Stage Compressor. Supposing the intake, or low-pressure, cylinder to be filled with free air just taken in, the advancing piston compresses the air until a point somewhat beyond half stroke is reached. At this point the delivery valves open, and during the remainder of the stroke the compressed air, at, say, thirty to thirty-five pounds pressure, is being forced out through the connecting pipe and passages into the second or high-pressure cylinder. Meanwhile, no work is being done by the high-pressure piston. On the return stroke the air at the low pressure which was delivered into the high-pressure cylinder is compressed to the required final tension and discharged. During this return stroke no work is done in the low-pressure cylinder, except that another charge of free air is drawn in. Thus, the intake stroke of the low-pressure cylinder is the compression and delivery stroke of the high-pressure, and *vice versa.* During the low-pressure intake stroke the portion of partly compressed air remaining in the pipe or passage connecting the cylinders is unaffected, as it is shut off from both cylinders by the valves at either end. At the beginning of the return stroke of the high-pressure cylinder the air in the connecting pipe begins to flow into this cylinder, and its pressure diminishes according to the relative volumes of pipe and cylinder. In the mean time the air is being compressed in the low-pressure cylinder, and when its tension exceeds that in the connecting pipe (that is, at, say, half stroke) it begins to pass through the delivery valves into this pipe. During the remainder of the stroke the low-pressure piston is in reality acting upon and compressing, not only the air in its own cylinder, but also that which is in the connecting pipe and high-pressure cylinder.

A serious disadvantage of the single-acting, two-stage compressor of this form is that the net resistances in the two cylinders are not equalized. Although the actual work of compression is designed to be the same in both cylinders, equalization of the resistances throughout both strokes is practically impossible because, in the second half of the forward stroke of the intake piston, the air delivered by it acts as a back pressure on the high-pressure piston, which is travelling in the same direction. This back pressure, in

turn, assists the movement of the low-pressure piston during its compression stroke. In this stroke, therefore, less total resistance is presented than during the compression stroke of the high-pressure piston. It has been pointed out by Mr. Frank Richards that, "to decrease the diameter of the high-pressure cylinder would tend toward equalization of the resistances, by allowing the intake cylinder to do more work, and compress the air to a higher pressure; but to raise the pressure (at delivery) in this cylinder would be to defeat the object of two-stage compression—that of allowing an efficient cooling of the air, and a reduction of its volume before its compression is too far advanced." In stage compression it is a fundamental principle that the cylinders should be so proportioned that the total work is divided equally between them. This secures the largest saving possible in the mechanical work of the compressor, as well as in efficiency of the cooling apparatus.

Double-Acting Two-Stage Compressors. The operation of this type is more satisfactory than that of the single-acting two-stage compressor, because, *first*, the cycle of operations during each forward and back stroke is the same; and, *second*, the distribution of the resistances throughout the stroke may be made more uniform.

A number of combinations in the arrangement of the steam and air cylinders are possible, but only three forms need to be noticed, as representing accepted practice, *viz.:* the straight-line, two-stage compressor (Figs. 7 to 12) and the duplex forms, consisting of a pair of cross-compound air cylinders, placed tandem to either twin-simple, or cross-compound steam cylinders (Fig. 14 to 23). The last-named is undoubtedly the best for large plants.

The principles of the mode of operation of all three designs may be illustrated by reference to Fig. 52, which shows diagrammatically a Norwalk two-stage straight-line compressor.

Assuming that the pistons have reached the end of their forward stroke, the conditions in the two cylinders are approximately as follows: The low-pressure cylinder (D) is full of air, practically at atmospheric pressure, while the high-pressure cylinder (G), together with the intercooler (F) and connecting passages, are oc-

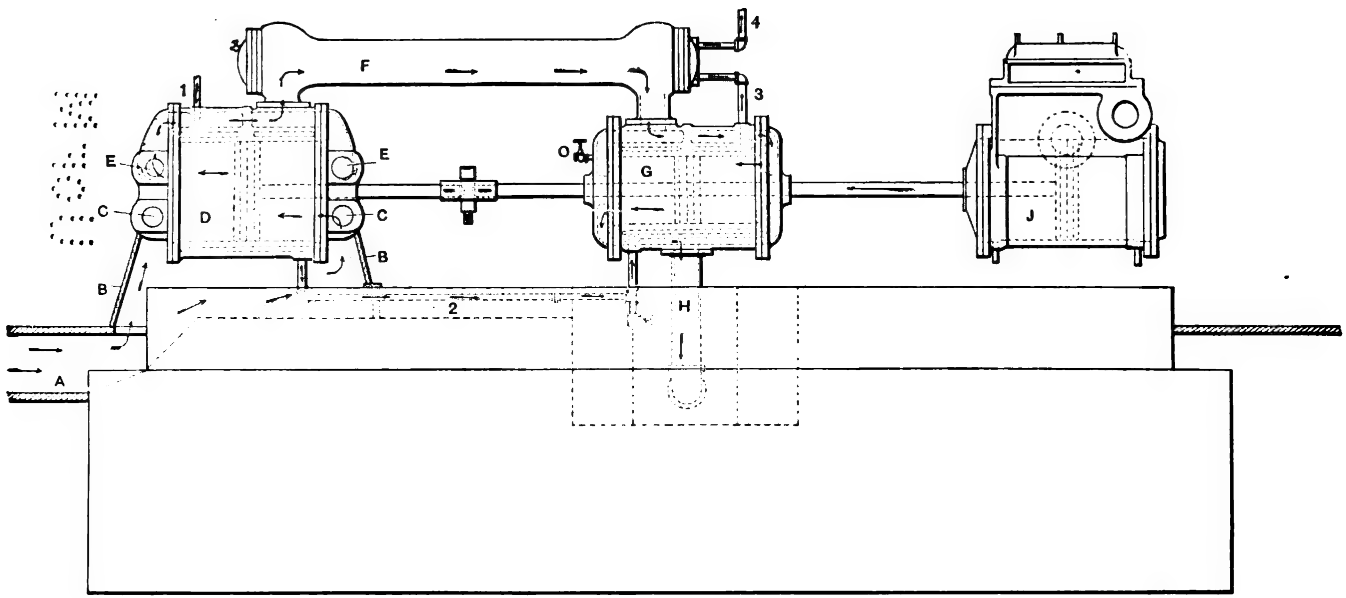


FIG. 52.—Diagram of Norwalk Two-Stage Compressor.

When pistons move as indicated by the arrow on the piston rod, the steam and air circulate in direction shown by arrows in the cylinders. Arrows on the water pipes show the direction of water circulation. A, inlet conduit for cold air; B, removable hoods of wood; C, inlet valve; D, intake cylinder; E, discharge valve; F, intercooler; G, high-pressure cylinder; H, discharge air pipe; J, steam cylinder; O, air relief valve, to effect easy starting after stopping with all pressure on pipes; 1, cold water pipe to cooling jacket; 2 and 3, water pipe; 4, water overflow or discharge.

cupied by air just delivered from the low-pressure cylinder, at, say, 30 to 35 pounds, or something less than one-half the final pressure. On the reverse stroke the free air in front of the low-pressure piston is compressed to 30 pounds and delivered into the intercooler and high-pressure cylinder, while the air already occupying the latter is brought up to the final pressure and discharged. This must be considered only as a rough description of what takes place in the air cylinders during a complete forward and back stroke.

As usually constructed for standard tandem, two-stage compressors, the volumetric capacities of the low- and high-pressure air cylinders are to each other in the ratio of about ten to four. The intention is to proportion the two cylinders so that their ratios of compression are nearly equal. Thus the distribution of work and the heat generated in the cylinders will be equalized and most effectually dealt with by the intercooler, provided the latter properly performs its functions. Practice as regards the relative volume of the intercooler and cylinders has not yet been completely standardized. It has undergone considerable change in the past few years. As clearer conceptions have been reached of the fundamentally important functions of the intercooler in stage compression, and in recognition of the fact that the first cost of even a very large intercooler is moderate, while its running expenses are practically nil, the tendency now is to make it of much greater volumetric capacity than formerly. Such increase of size produces substantial gain in thermodynamic efficiency. The hot compressed air delivered by the low-pressure cylinder is kept longer in contact with the cooling surfaces because of its reduced speed of flow through the larger cross-sectional area of the intercooler, and it enters the high-pressure cylinder, to undergo the second stage of compression, with a temperature that may readily be made to approximate closely to the normal. On the other hand, it is clear that the connecting passages between the cylinders and intercooler should be of as small volume as is consistent with freedom from excessive frictional resistance in the flow of the air through them; because the air occupying these passages at any given time is exposed to but little cooling save that due to radiation.

With these points in view, it may be assumed in good practice that, if the volume of the low-pressure cylinder be taken as 10, then the volume of its connection with the intercooler should be, say, 1.5, of the intercooler 4, of the connection to the high-pressure cylinder 1.5, and of the high-pressure cylinder 4. (It may be noted that there is no reason why the net capacity of the intercooler should not be even greater than is here assumed.) Having these proportionate volumetric capacities, the following sequence of operations will take place while the compressor is making a single stroke. Suppose this stroke to be from right to left, as indicated by the arrows in Fig. 52.

By the previous stroke (from left to right) the intercooler and both of its connections to the cylinders, representing a volume $= 1.5 + 4 + 1.5$, were filled with air compressed, at, say thirty pounds. This body of air was then shut off from both cylinders by their respective valves, and has lost part of its heat and pressure by the action of the intercooler. After reversal, and during the first part of the following (left-hand) stroke, the low-pressure piston acts only on the cylinderful of free air just taken in (volume$=10$).* While this is being compressed, the advance of the high-pressure piston causes the compressed air already in the intercooler and its connections to begin to flow into the high-pressure cylinder, thereby increasing in volume and decreasing in pressure, until a point, say, a little beyond mid-stroke is reached. On passing this point the air pressure in front of the low-pressure piston rises slightly higher than that in the intercooler and the corresponding low-pressure delivery valves open, so that the low-pressure piston acts upon the entire body of air—volume $= \frac{10}{2} + 1.5 + 4 + 1.5 + \frac{4}{2} = 14$. Then, until the end of the stroke, both cylinders are in communication through the intercooler, *i.e.*, from the left-hand end of the low-pressure cylinder to the right-hand end of the high-pressure cylin-

* The general method of analysis here given is similar to that employed some years ago by Frank Richards, "Compressed Air," pp. 86–87, though the quantities used are taken to represent a closer approach to current practice in the proportions of the parts.

der, as shown by the arrows in the cut, and an approximate equalization of pressure is established throughout.

Up to the time of the opening of the left-hand, low-pressure delivery valves, the air in the intercooler, and still under its influence, has been isolated from the low-pressure cylinder, in which compression has progressed without other cooling than that effected by the cylinder water-jacket. But when the warm, partly compressed air begins to pass from the low-pressure into the high-pressure cylinder, through the intercooler, the influence of the latter is exerted upon a new body of air. At the end of the left-hand stroke the closing of the delivery valves again shuts off the air in the intercooler from both cylinders. The high-pressure cylinder, on the right-hand side of the piston, is occupied by a body of air whose temperature has been reduced by the combined effect of the intercooler and both water-jackets to a point much below that due to the working pressure of the low-pressure cylinder, and whose pressure has dropped correspondingly.

Now, in the latter part of the left-hand stroke, when the low pressure delivery valves have opened and the piston of this cylinder is acting on the volume 14, as stated above, a portion of this air (volume $= \frac{2+1.5}{14} = 25$ per cent.) of the total has passed beyond the influence of the intercooler, and another portion (volume $= \frac{5+1.5}{14} = 46$ per cent.) has not yet reached it. A similar statement of the distribution of the air with respect to the intercooler may be made for other points of the stroke. At the end of the left-hand stroke under consideration the volume of compressed air in the low-pressure cylinder $= 0$, in the intercooler and its connections $1.5 + 4 + 1.5 = 7$, and in the high-pressure cylinder 4, a total of 11, of which 1.5 has not reached the intercooler but has been affected only by the low-pressure water-jacket.

This analysis should be clearly understood in forming a correct estimate of the work actually accomplished by the intercooler. It emphasizes the importance not only of employing an intercooler whose volumetric capacity is large relatively to the cylinders,

but also of making the connecting passages small. It is evident that one-half of the total work of compression—that performed in the high-pressure cylinder—is done solely under such cooling influence as may be exerted by the water-jackets of this cylinder. The jackets of both cylinders should, therefore, be as large in area as possible, with an efficient circulation of cold water. They should cover not merely the cylinder barrels, but as much of the heads as the spaces occupied by the valves will permit. In the latter respect some recent compressor designs are deficient.

The details of the distribution of the air in the foregoing description apply exactly only to compressors in which the air cylinders are tandem to each other. In the duplex stage-compressors, where the air cylinders usually are, and always should be, cross-compounded, the cycle of operations is different because the pistons, instead of moving together in the same direction, work with one crank 90° in advance of the other.

As stated above, it is intended in stage compression that the total work done shall be equally divided between the air cylinders. But, by reason of the frequent variations in receiver pressure, upon which depends the actual terminal pressure of the high-pressure cylinder, an approximate equalization only can be attained in practice. On the basis of some terminal pressure taken as normal, such diameters are assigned to the cylinders as will make their compression ratios equal, or nearly so. Take, for example, a pair of cylinders, 15 ins. and 24 ins. in diameter, to produce a final pressure of 85 lbs. gauge. Assuming that the air between the stages is cooled to the original temperature, the absolute intake pressures of the cylinders will be inversely proportional to the squares of their diameters, or: $15^2 : 24^2 :: 14.7 : 37.64$. The absolute pressure of 37.64 lbs., as delivered by the low-pressure cylinder, is theoretically equal to the intake pressure of the high-pressure cylinder. The ratio of compression in the low-pressure cylinder is: $\frac{14.7}{37.64} = 0.3905$; and in the high-pressure cylinder: $\frac{37.64}{99.7} = 0.3775$. This would be quite as close to perfect equalization as is necessary.

Construction of the Intercooler. A number of forms are now in use. As commonly constructed for straight-line compressors, the intercooler consists of a long cylindrical chamber, containing a number of parallel, thin brass (sometimes wrought-iron) tubes, through which cold water is circulated. The air to be cooled passes through the spaces between the tubes. The intercooler is placed in a convenient position between and above the cylinders, and as close to them as possible, so that the connecting passages may be short and of small volume. As already stated, the air contained in these passages at any given time is denied the cooling

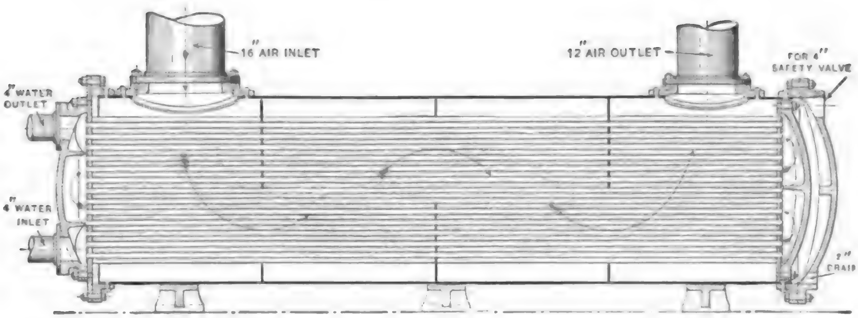


FIG. 53.—Horizontal Intercooler. Ingersoll-Rand Co.

effect both of the cylinder water-jackets and of the intercooler itself. In Fig. 52 the intercooler is indicated at F: in Fig. 8 the Norwalk intercooler is shown in longitudinal section. Fig. 53 illustrates a large horizontal intercooler, as built by the Ingersoll-Rand Co. Another design, for cross-compound air cylinders, by the Sullivan Machinery Co., is shown in Fig. 54, a large intercooler being placed crosswise below the cylinders. In many cross-compound compressors the intercooler is mounted above the cylinders. The tendency now is to increase the size and volume of the intercooling chamber, relatively to the volume of the cylinders.

The air delivered from the low-pressure cylinder passes on its way to the high-pressure cylinder between the intercooler tubes, which must be sufficiently close together thoroughly to split up the body of air traversing the intermediate spaces and so secure the maximum cooling effect. It is intended that the temperature of

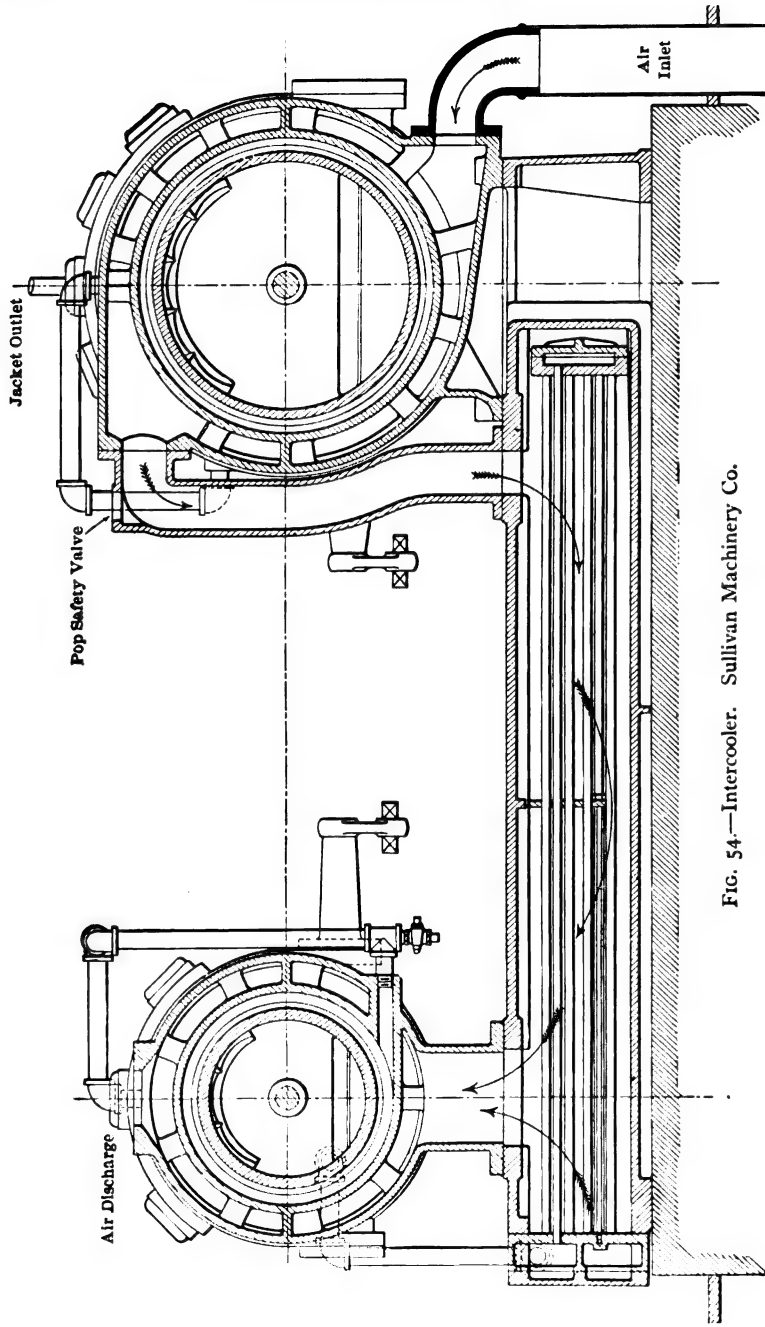


Fig. 54.—Intercooler. Sullivan Machinery Co.

the air, on leaving the intercooler and entering the high-pressure cylinder, shall be reduced nearly to the normal. The effect of this drop in temperature upon the compression curve of a two-stage compressor is shown by Fig. 57; the curve of the high-pressure cylinder should, and often does, begin close to the isothermal line.

In the construction of the intercooler brass tubes are perhaps preferable to those of iron because of their higher conductivity; but, on the other hand, iron tubes cost less, and on account of their greater roughness present a larger cooling surface to the air flowing between them. In either case they should be as thin as is consistent with the necessary strength. The tubes are expanded into tube-sheets at each end, and by means of two or more baffle-plates, set equidistant between the ends, the air is compelled to pass through the entire volume of the intercooler. The water-heads at the ends are so divided that the water is caused to circulate actively back and forth several times, before it is finally discharged, as shown by the small arrows in Fig. 53. For convenience the water supply is usually connected with the circulating system of the cylinder-jackets.

Fig. 55 illustrates a peculiar system of intercooling adopted in the Leyner compressor. A number of horizontal iron or bronze tubes are enclosed in the annular water-jacket spaces, between the inner and outer shells of the cylinder. The piston being at the middle point of its stroke, the inlet valves at the left-hand end of the low-pressure cylinder are open and taking in air. Meantime the air in front of the piston, having been compressed, is passing out through the delivery valves into the air chamber or head at the right-hand end of the cylinder. This air is thence forced by horizontal baffle-plates in the air chamber through the upper set of intercooler tubes, and into the left-hand end of the cylinder. It flows next to the right, through the lower set of intercooler tubes, and as shown by the arrows enters the lower tubes of the high-pressure cylinder. From the right-hand air head of this cylinder the air is directed by baffle-plates back through the upper set of tubes to the left-hand end of the high-pressure cylinder, into which it enters

through the corresponding inlet valves. The air already compressed in this cylinder is shown as passing through the large upper aftercooling tubes to its own air chamber, which leads to the discharge pipe. It will be noted that the low-pressure air, in being subdivided into small volumes and compelled to change its direction several times in passing back and forth through the intercooler tubes, is well cooled before entering the high-pressure cylinder. It is important that the copper tubes of the intercooler be kept clean. As the oil carried over by the air tends to deposit on the tubes, they should be so arranged as to be readily accessible for cleaning. The intercooler of the Schram (English) two-stage compressor is a vertical chamber, also filled with small tubing. The water enters at the bottom, passes up through one-half of the tubes and down through the other half, the lower water-head being divided accordingly. The air from the low-pressure cylinder enters at the top of the intercooler, passing out at the bottom into the high-pressure cylinder.

Although the relatively small intercoolers of ordinary two-stage compressors are imperfect in their action, as has been pointed out, it is nevertheless possible to attain a high degree of efficiency from intercoolers of large capacity. A well-known example may be cited: the plant of the Paris Pneumatic Supply Co., in which Riedler two-stage compressors are used. Spray injection is applied to both cylinders, and also a plain intermediate receiver of very large capacity, but without tubes. The air is compressed to 88 pounds, and the indicator diagrams of the air cylinders exceed in area the true isothermal diagram by only 12.07 per cent.* That is, the work done twice is about 12 per cent. of the total work, the total efficiency having the high value of 77 per cent.

To show the results obtained by thorough cooling of the air between the cylinders, a comparison of the work done by single- and double-stage compression may be made. Frictional losses will be omitted in each case, and no account will be taken of the cooling due to the cylinder water-jackets.

1. A single-stage compressor, producing a gauge pressure of

* *Proceedings Institution of Civil Engineers*, London, Vol. CV, p. 180.

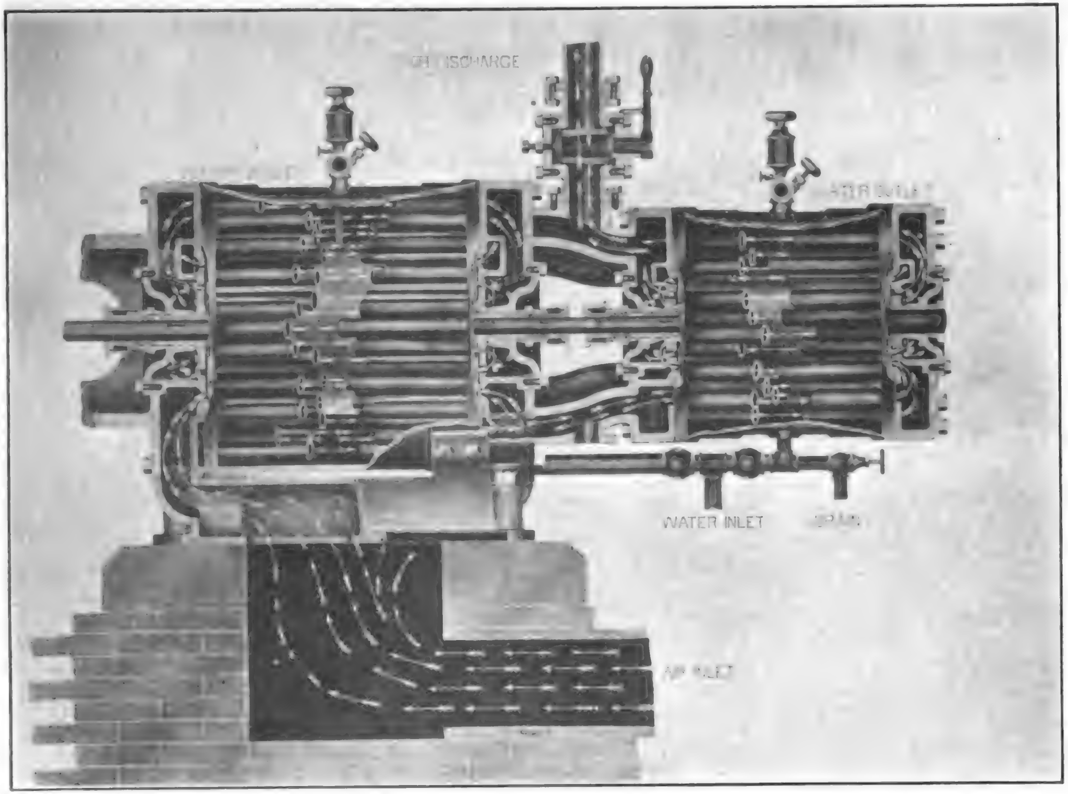


Fig. 55.—Leyner System of Intercooling.

70 pounds at sea-level, with a 24-inch cylinder and a piston speed of 400 feet per minute, will have a capacity in terms of free air at normal temperature of 1,256 cubic feet per minute. For adiabatic compression, the mean cylinder pressure will be 33.83 pounds and the horse-power 184.38.

2. For doing the same work in a two-stage compressor, provided with an intercooler capable of reducing the temperature of the air to the normal between the cylinders, it may be assumed that the low-pressure or intake cylinder has the same diameter, 24 inches, and that the pressure produced in it is 35 pounds. The mean pressure (adiabatic), corresponding to 35 pounds terminal pressure, is 21.6 pounds, and the horse-power 118.19. The diameter of the high-pressure cylinder, under the assumed conditions, is found by making the piston area inversely proportional to the increase in absolute pressure of the air delivered to it by the low-pressure cylinder, *i.e.*, in the ratio of $14.7 : 35 + 14.7 = 1 : 3.38$. This gives an area of 135 square inches, equivalent to 13 inches diameter. Compressing in this cylinder from 35 to 70 pounds gauge, the mean effective pressure will be 28.74 pounds, and the horse-power, 46; or a total for both cylinders of $118.19 + 46 = 164.19$ horse-power.

Compared with the power required for doing the same work in the single cylinder, this shows a saving of: $184.38 - 164.19 = 20.19$ horse-power, or about eleven per cent. The theoretically perfect cooling between the cylinders here assumed would not be attained in ordinary practice, however, and the frictional loss in the stage compressor would probably be a little greater than in the single-cylinder machine; so that the net gain due to inter-cooling may in this case be taken at, say, seven to eight per cent. The saving is considerably increased in dealing with higher pressures. (For "Stage Compression at High Altitudes," see p. 219.)

The advance made in recent years in the design of intercoolers is further illustrated by Fig. 56, showing a new design of the Ingersoll-Rand Co. It is provided with pipe connections for draining off the water deposited as a result of the reduction in temperature. These coolers may be employed also as "receiver-after-coolers," which are now considered as almost essential adjuncts

of well-installed large plants. (See Chapter XI.) A similar appliance may be employed advantageously as an ante-cooler for the intake air.

The useful effect of small intercoolers, such as are frequently mounted above the cylinders of straight-line compressors, should

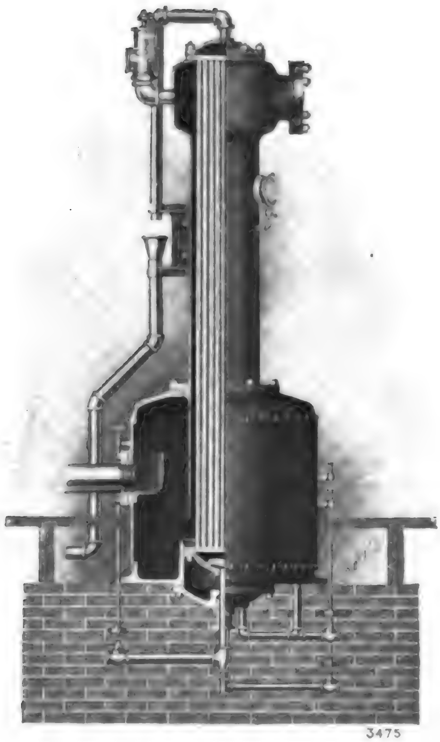


FIG. 56.—Vertical Intercooler. Ingersoll-Rand Co.

not be misunderstood nor exaggerated. It must be remembered that the best economy in air compression is obtained only by cooling *during* compression and before the air leaves the cylinder. Hence, in addition to the intercooler, the largest possible water-jacket area should be provided.

The relation between the compression curves of a two-stage

compressor is shown in Fig. 57, the adiabatic and isothermal curves being also laid down.* These cards, not accurately reproduced here, were taken from a pair of cylinders measuring 7½ and 14 × 16 inches, compressing to 110 pounds gauge, at 135 revolutions per minute, or 360 feet piston speed. Initial temperature of cooling water, 55°; temperature at discharge from jackets and intercooler,

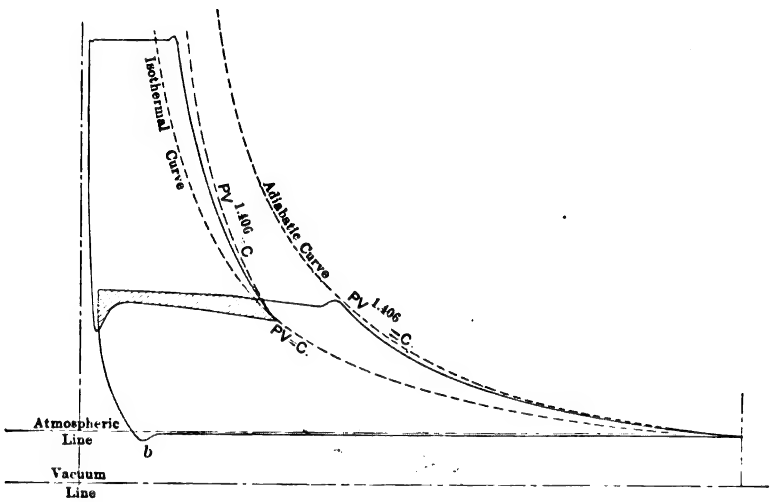


FIG. 57.—Combined Air Card of Two-Stage Compressor.

62° F. Several points are to be noted in connection with these combined two-stage cards:

First. The overlapping of the high- and low-pressure cards indicates a loss, because the work represented by the area of the overlap is in reality *work done twice.* This is the result of the drop in pressure between the cylinders, which is caused by the resistance presented by the discharge valves of the low-pressure and the inlet valves of the high-pressure cylinder, together with the friction in the air passages and intercooler. While this loss is unavoidable,

* This combined indicator card, which does not show all the minor irregularities in the lines, is from a Rand cross-compound compressor. It accompanies an article by F. A. Halsey, on "The Analysis of Air Compressor Indicator Diagrams," *American Machinist*, March 3d, 1898, p. 158, and is reproduced here by permission.

it should be reduced as much as possible by making the valves, ports, and connecting passages of ample size.

Second. As with single-cylinder dry compressors, the compression lines of the individual cylinders of most stage compressors depart but little from the adiabatic curve. Aside from the thermodynamic advantage of dividing the total compression between two or more cylinders, and thereby lowering the average and final temperatures, it is the intercooler that must be relied on for furnishing the chief element in economical working. By its abstraction of heat the volume of air entering the second cylinder is reduced, so that $PV^{n=1.4}$ = constant becomes approximately $PV=C$, on beginning the second stage of compression. But the compression line again rises rapidly from this point and continues not far below the adiabatic.

Indicator cards from dry compressors which do not show approximately this relation between the lines are always open to suspicion. A leaky piston, for example, will lower the compression curve and make it appear that much better work is being done than is really the case. It may be observed that, other things being equal, a lower curve is often obtainable from a small than from a large compressor, because the ratio of area of water-jacket to the volume of the cylinder is greater.

In constructing and reading a combined indicator card from both cylinders of a stage compressor (like that shown in Fig. 57), the adiabatic line applying to the compression in the second cylinder should be represented in its proper place. The complete graphic relation between the several heat curves is thus set forth.

Third. It is an advantage of stage compression that there is practically but one clearance space—that in the low-pressure cylinder—and, as the air in this cylinder is at a low pressure, the resulting reduction in net volumetric capacity is moderate, for it is evident that the loss due to clearance is proportionately less for low than for high pressures. The piston clearance of the high-pressure cylinder cannot affect the volume of air delivered, because all the air discharged from the low-pressure cylinder goes to the high-pressure and, barring leakage, must pass through it.

8

The heating of the cylinder walls and pistons reduces somewhat the working volumetric capacity of an air compressor because, as the entering air is warmed, a smaller weight of it is taken into the cylinder at each stroke. Although the degree of this heating cannot be formulated, it is obvious that it is less in a two-stage than in a single-cylinder compressor; for, aside from the effect of the intercooler, the smaller quantity of heat developed in each cylinder is more efficiently dealt with by their respective water-jackets.

CHAPTER VII

AIR INLET VALVES*

THE proper design and working of the inlet or suction valves exert an important influence on the efficiency of the compressor, and perhaps no other one portion of air-compressor mechanism has received so much attention. Nevertheless, that there are still wide differences of opinion as to the best design for inlet valves is evidenced by the great variety of types used by compressor-builders and the lack of clearly defined distinctions as to their applicability under different working conditions. Reference to almost any compressor catalogue will show that the purchaser has a choice of several types, with but little to guide him in making a selection.

In the older forms of wet compressor various patterns of clack-valve were employed, as exemplified in the Dubois-François compressor. Though not now used in this country, they have by no means been abandoned in Europe; witness the Guttermuth valve and the elaborate, cam-controlled clack-valves of some large compressors built by Schneider & Co., Creusot, France. For years poppet valves of numerous types held the field in the United States almost exclusively. They are furnished with springs, and are usually actuated solely by difference of air pressure; though in a few designs mechanically controlled poppets were introduced, such as those of the old Rand mechanical valve-gear and others, examples of which are still occasionally to be found in use. While poppet valves have continued in favor for certain kinds of service, and are likely to remain so, many other forms of inlet valve have been

* This chapter is devoted chiefly to spring poppet valves and others which operate by difference of air pressure. For discussion of those inlet valves whose movements are under mechanical control, see Chapter IX.

successfully applied in the course of the development of the modern compressor. Modifications of the Corliss rotary steam valve, first used in the Norwalk compressor, have now been adopted in compressors of many other makes, such as the Nordberg, Sullivan, Laidlaw-Dunn-Gordon, and Allis-Chalmers. There are at least two inlet valves which cannot be included in any of the other classes, *viz.:* the Sturgeon valve, placed in the cylinder head and operated by frictional contact with the piston rod, and the ingenious Ingersoll-Sergeant piston inlet, which opens and closes by its own inertia at the end of each stroke. Both of these operate under fixed conditions, independently of differences in air pressure within and without the cylinder.

The two chief requisites of all inlet valves are: 1. That they shall have a sufficient area of opening to permit free entrance of the air. 2. That they shall open readily near the beginning of the stroke, with a minimum of resistance, remain open until the end of the stroke, and then close promptly.

There are several questions affecting the design and operation of the usual types of inlet valve, which are closely related to the working of the air cylinder itself. The point of the stroke at which the inlet opens should depend on the piston clearance and the air pressure under which the compressor is working. Spring-controlled valves, or those operated mechanically, are sometimes incorrectly designed or set, so as to open exactly at the beginning of the stroke or a fraction later; in which case the clearance air is first exhausted through the valves and then, as the piston advances, the outside air begins to enter. This being so, it is evident that no clearance at all would be shown on the indicator card.

As already pointed out, although piston clearance causes a reduction in volumetric capacity of the cylinder, it not only does not involve a corresponding loss of work, but is in reality beneficial, in assisting to overcome the inertia of the reciprocating parts of the compressor. A large part of the work expended in compressing the clearance air is thus recovered. But when the clearance air is exhausted wholly or in part by a premature opening of the inlet valves, the work represented by it is lost. With spring-controlled

poppet valves the proper adjustment is a question of the strength of the spring, and since the effect of clearance varies with the air pressure, the valves must be regulated for the pressure carried in each particular case. Any exhaust through the inlet valves is readily detected by the noise. When they are properly set, the compressor works more smoothly and the power consumed is slightly reduced. On the other hand, if the valves open too late in the stroke—due, for example, to a temporary reduction in working pressure—a little more power is required, this condition being shown by the slight drop in the re-expansion line at the point *b* (Figs. 48 and 57).

For inlet valves which are opened and closed mechanically, an adjustment to the working conditions is even more imperative than in the case of valves controlled only by springs. If incorrectly set or timed with respect to the stroke of the piston, they may be forcibly opened too early in the stroke or closed before the end. Premature closing obviously reduces the volume of intake air, and with it the volumetric capacity of the compressor. Its effect on the indicator card is to lower the compression line near the beginning of the stroke, so as to approach the isothermal curve and make it appear that the compressor is doing abnormally good work.

The total area of the inlet ports varies greatly in compressors of different makers. It is sometimes as small as 4 or 5 per cent. of the piston area, running up to a probable maximum of 12 to 15 per cent. As the proper area is really a function of the piston speed, it may be made less for slow- than for high-speed compressors. However, in one of the Leyner 2-stage compressors, with a 22-inch low-pressure cylinder and running at the moderate piston speed of 390 feet, the intake port area is 14.2 per cent. of the piston area. (The Leyner valves are of a special type, described hereafter.) To insure freedom from excessive frictional resistance against the inflow of air, the inlet area, under average conditions and for ordinary forms of valve, should be not less than, say, 8 or 10 per cent. of the piston area. But extremes should be avoided. If poppet valves are made unnecessarily large, their inertia becomes great; and if too numerous, there are not only more parts to care

for, but valuable water-jacket area on the cylinder heads must be sacrificed.

In the two-stage, straight-line, "Hurricane-inlet" compressors, of the Ingersoll-Rand Co., type AA–2, for cylinders from 15-in. to 24-in. diameter, the inlet area of the low-pressure or intake cylinder averages 13.2 per cent. of the piston area. For the high pressure cylinders of the same compressors, the poppet valves have an average inlet area of about 11 per cent. The inlet area of the duplex, two-stage compressors, type O–2 of the same builders, averages 13.6 per cent. of the piston area for both low- and high-pressure cylinders.

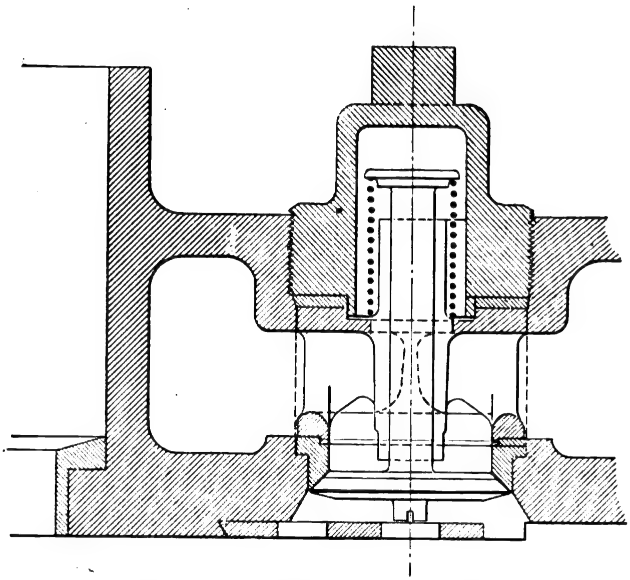


FIG. 58.—Norwalk Poppet Inlet Valve.

In the two-stage compressors of the Laidlaw-Dunn-Gordon Co. the percentage is from 12 to 14.

Poppet Inlet Valves. One of the commonest forms is the mushroom valve, two types of which are shown in Figs. 58 and 59. While the total inlet area should be ample, there are two special requirements in the case of ordinary poppet valves: (1) the area of each individual valve must be moderate, or the valve will become

too heavy, causing unnecessary injury to the valve seat, and by its inertia too great a resistance to the control of the spring; (2) the lift must be small, in order to attain prompt opening and

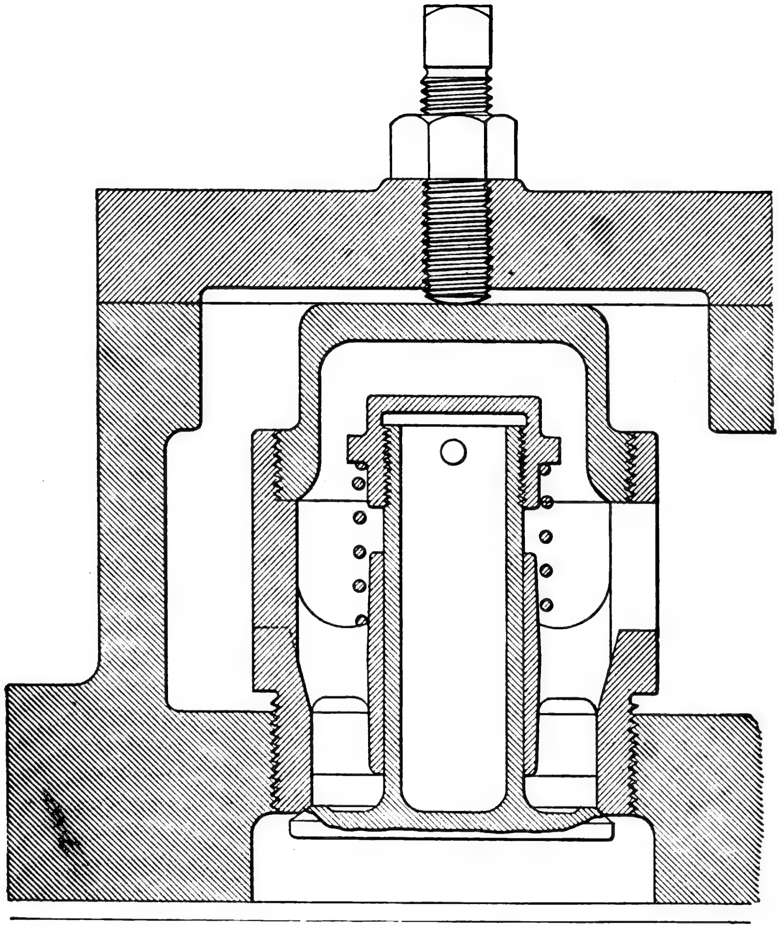


FIG. 59.—Laidlaw-Dunn-Gordon Poppet Inlet Valve.

closure, and to reduce "chattering," as well as wear. For these reasons the total area required is furnished by a number of independent valves, generally from four to six, which are set in each cylinder head.

The valve is of steel or bronze, with an easily removable bronze seat, the contact surfaces being ground true and the seating often coned. To control and close the valve promptly its stem is provided with a spiral spring. The stem works in guides, forming part of the seat and valve casing, which is screwed into the cylinder head so as to be readily removed when necessary for adjustment or repairs. Brass springs are used, to avoid the effects of corrosion, and must be easily compressible to allow the valve to open freely under a small difference of pressure; that is, as early in the stroke as possible after the clearance air has re-expanded. The springs should be made of the best material and accurately proportioned to present no more than the minimum requisite resistance to opening. Under actual working conditions the pressure of the springs varies from, say, three ounces to eight or even ten ounces per square inch of valve area.

Ordinary poppet valves are opened by the atmospheric pressure from without, when a certain degree of rarefaction of the air inside the cylinder has been produced by the movement of the piston; in other words, when the difference of pressure, after the clearance air has re-expanded, becomes sufficient to overcome the resistance of the spring, and compress it. In accomplishing this the piston must advance some distance before any air can enter the cylinder. The loss of volumetric capacity thus caused, in terms of free air, is probably rarely less than two to three per cent., and is often more. At sea-level a spring pressure of five ounces per square inch of valve area causes a loss of about two per cent. The diagram, Fig. 60,* shows the effect of spring resistance in reducing the volumetric capacity of a compressor at different altitudes, from sea-level to 15,000 feet elevation.

With spring-controlled poppets there is more or less irregularity in the entrance of the air, because, while the pressure of the outside air tries to open the valve, the action of the spring tends to keep it closed. This often produces "chattering" or "dancing" of the valves, and has led among other things to the introduction of various mechanical devices for definitely controlling them, as will

* Reproduced by permission from *Engineering News*, May 30th, 1901, p. 391.

be noted later. As the springs lose their original elasticity, and undergo alterations in strength, they require regulation from time to time; outside adjusting nuts on the valve stems may be provided

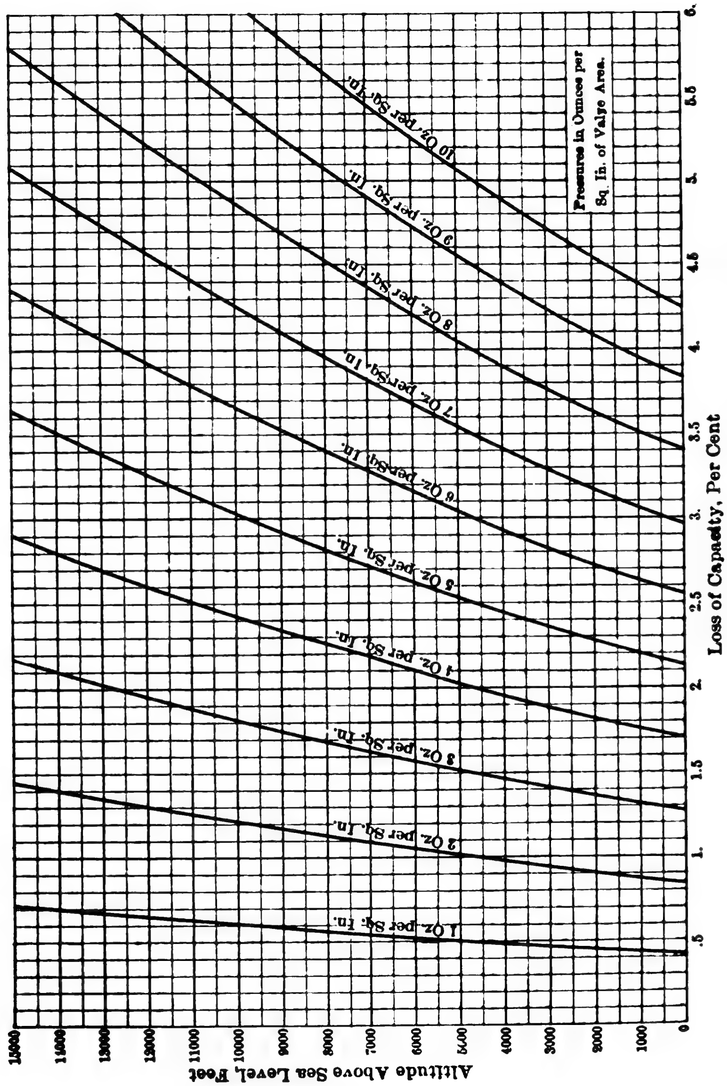


Fig. 60.—Effect of Valve Spring Resistance on Volumetric Capacity of Compressor.

for this purpose. If the springs be too slack, the chattering increases; if too tight, the valves will open late in the stroke, and the intake air occupying the cylinder will have a density less than that of the atmosphere. But, aside from spring resistance, the rate of inflow of the intake air is variable. This is due to the variation in speed of the piston. When its speed is greatest, at the middle of the

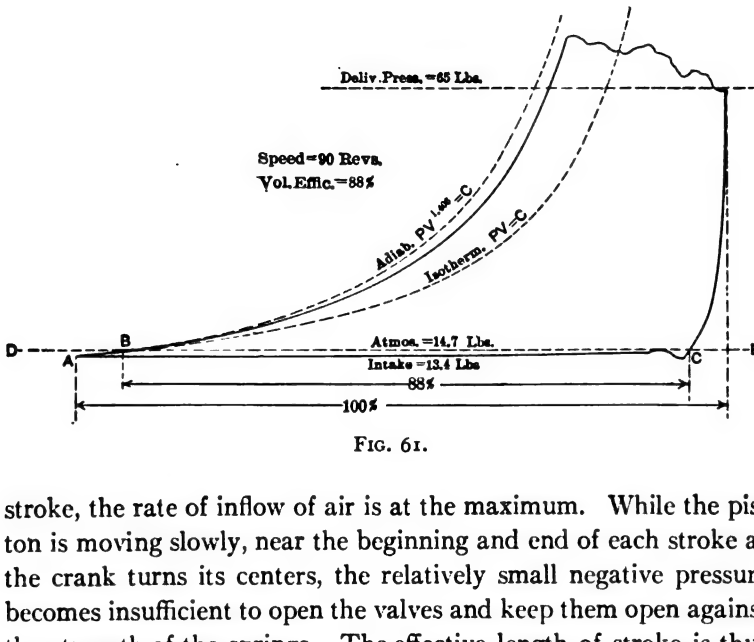


FIG. 61.

stroke, the rate of inflow of air is at the maximum. While the piston is moving slowly, near the beginning and end of each stroke as the crank turns its centers, the relatively small negative pressure becomes insufficient to open the valves and keep them open against the strength of the springs. The effective length of stroke is thus shortened.

The total valve resistance, including that due to throttling of the intake air and friction in passing through the ports, must be kept as small as practicable, but can never be entirely eliminated. With some forms of inlet valves, other than spring poppets, the resistance becomes very small, and sometimes almost inappreciable. Its usual effect is shown on the diagram, Fig. 61. There is generally sufficient resistance to keep the admission line, A C, at an appreciable distance below the atmospheric line, D E, throughout the stroke; the amount of loss from this cause being measured by the

area of the indicator diagram lying below the atmospheric line. If the inlet area be too small or the valves poorly designed, the resulting negative pressure may amount to one or two pounds per square inch. The point B, where the compression line crosses the atmospheric line, is the point of the stroke which must be reached by the piston before any useful work is done, and the volume passed through in travelling from A to B represents the loss in volumetric capacity from this cause. The total loss of volumetric capacity, including that due to piston clearance, is represented by the length of A B + C E, and the volumetric efficiency of the compressor is measured by the length of the line B C, projected on the atmospheric line.

Notwithstanding certain inherent disadvantages, the poppet valve in different forms is widely used, for both inlet and discharge. It is simple in construction, easily regulated, and in case of leakage, due to cutting and unequal wear of the seating surfaces, is readily removed and re-ground. In stage compressors it is often used for the high-pressure cylinder, even when some other type is preferred for the low-pressure. Poppet inlet valves not infrequently cause trouble by sticking in their seats on account of the accumulation of gummy oil. Or, they are sometimes clogged by deposit of carbonaceous matter from decomposition of the lubricant, produced by excessive heating of the cylinder. The valves should be kept clean, and are therefore designed to permit ready access.

One of the recent forms of Norwalk two-stage compressor has a special poppet inlet valve, designed for use when it is desired to employ air at two different pressures, obtained from a single compressor. In stage compression, though the air is actually produced at two pressures, of, say, 25 to 30 and 80 to 100 pounds, respectively in the low- and high-pressure cylinders, yet, if a part of the volume delivered by the low-pressure cylinder be drawn from the intercooler, the high-pressure cylinder fails to work satisfactorily. The air remaining in the intercooler expands to a lower pressure before going to the high-pressure cylinder, so that the ratio of compression in this cylinder is increased, and the heat gener-

ated is raised to a correspondingly higher degree. With such a rise in temperature as would be produced by increasing the ratio of

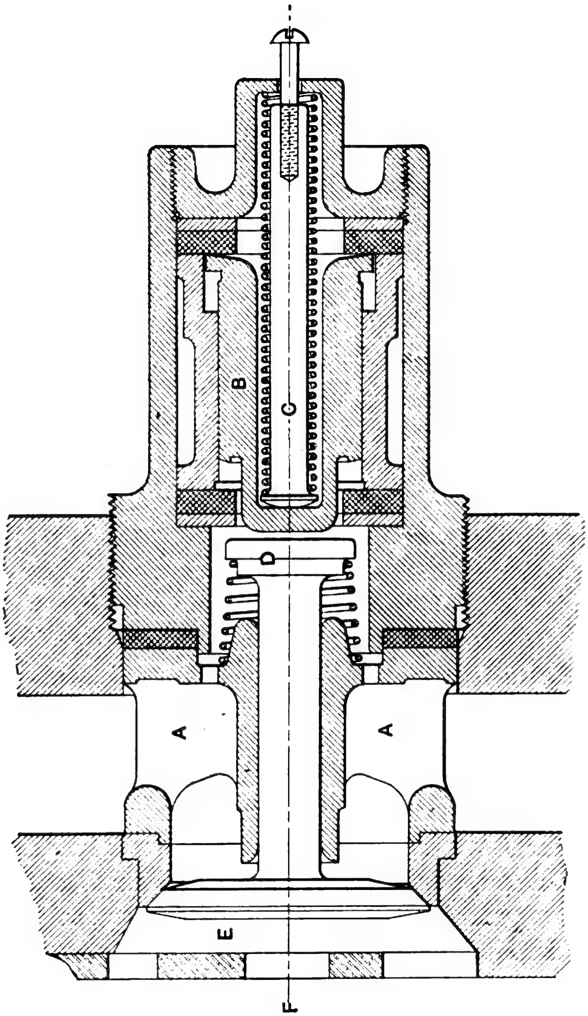


FIG. 62.—Skip-Valve. Norwalk Iron Works Co.

compression from, say, three to fifteen or twenty, proper lubrication is impossible, and the conditions would be favorable for an explosion in the cylinder.

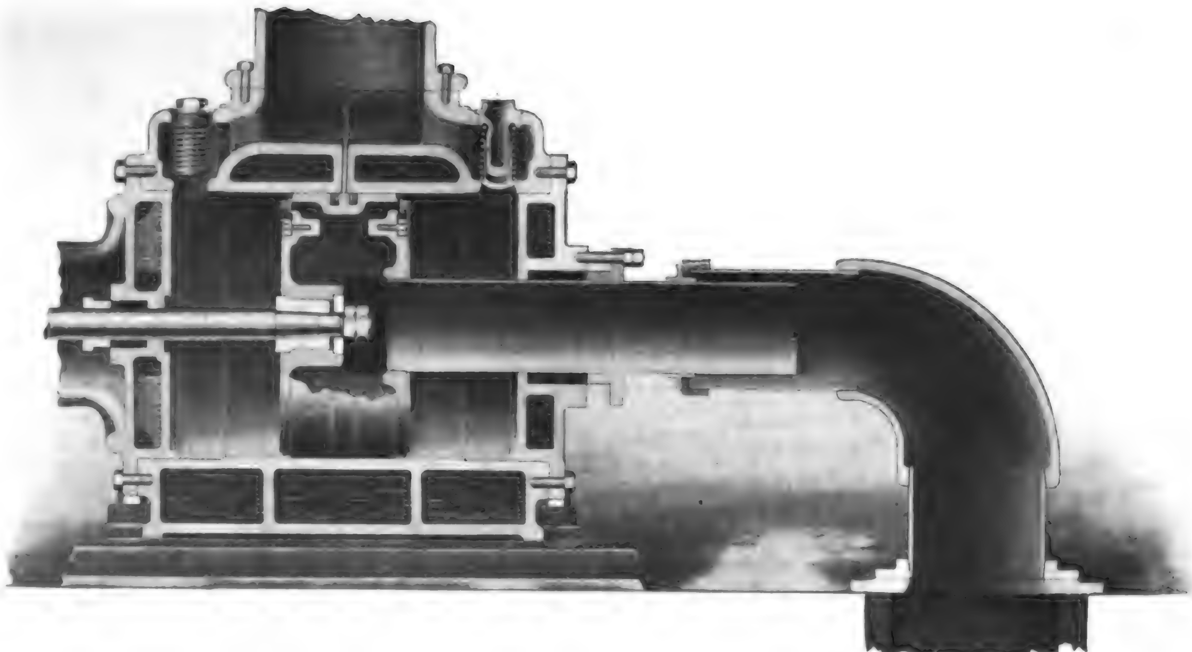


FIG. 63.—Ingersoll-Rand "Hurricane Inlet" Valve. Section of Cylinder and Piston, Showing Valves in Position.

This difficulty is met by using "skip-valves" (Fig. 62) as inlet valves of the high-pressure cylinder. They are designed to open, and remain open, whenever the high-pressure inlet air falls below the normal, by reason of having drawn off a portion of the air from the intercooler. The high-pressure cylinder is thus temporarily unloaded in part, since the air entering at each stroke is returned to the intercooler. The skip-valve is a mushroom spring-poppet, D E, carried in the guides A, A. Above the valve is a small, spring-controlled plunger B, the space below which is occupied by air at intercooler pressure. When this pressure falls below that for which the spring C is set, the plunger advances and forces open the inlet valve, holding it open until the intercooler pressure rises sufficiently to cause the plunger to recede. The valve is then free to work automatically in the usual manner. The action of the valve thus adjusts itself constantly to the varying pressure of the intake air coming from the intercooler; and the variation in consumption of power by the high-pressure cylinder is taken care of by the governor applied to the steam end of the compressor.

Ingersoll-Rand "Hurricane Inlet" Valve. In many of the Ingersoll-Rand compressors, the inlet valves are placed in the piston. Fig. 63 shows a longitudinal section through the cylinder and piston; Fig. 64, an enlarged section of the piston and valves. The piston is hollow and into its rear end is screwed a hollow back piston-rod, for admitting the air. The pipe is provided with an ordinary stuffing box in the cylinder head. There are two large, ring-shaped valves (one in each face of the piston), of T cross-section and made of oil-treated, annealed steel. The valves are a little smaller in diameter than the piston and are held in place, without springs or other connection, by a guide-plate bolted to the piston face. Their play is limited by these guide plates (see Fig. 64), which contain a series of circular ports, furnishing additional area for the passage of the air. The valves are readily taken out for regrinding when necessary, by removing the guide plates which are attached to the piston by tap-bolts.

While the compressor is running, the air is drawn in through the hollow piston-rod in practically a constant stream, passing

through either valve, first into one end of the cylinder and then into the other. At the beginning of each stroke the valves are alternately opened and closed by their own inertia, as the piston reverses its motion. The valve in that face of the piston which is toward the direction of movement is always closed, while the other is open for the passage of the air entering through the hollow rod into the cylinder behind the piston. On account of the large di-

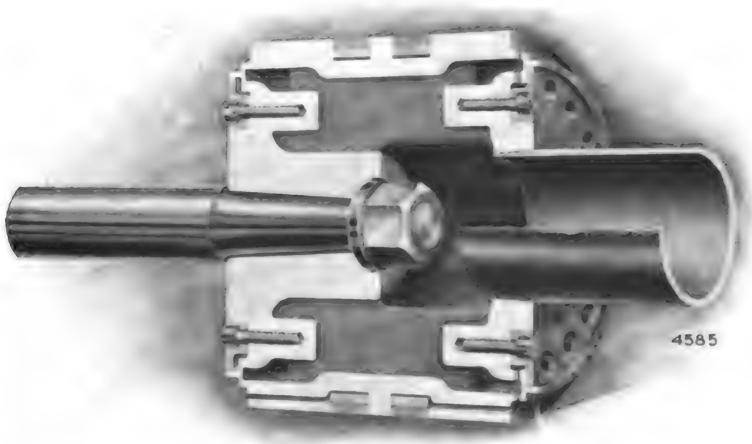


FIG. 64.—Ingersoll-Rand "Hurricane Inlet," Enlarged Section.

ameter of the valves their throw, or lift, is small—in ordinary compressors say three-eighths inch.

In the "Hurricane-Inlet" cylinders the area of the hollow piston rod, or inlet tube, is usually 13 to 14 per cent. of the piston area. Though the actual port area of the valve is less than this —say 8 per cent.—the velocity of flow is moderate. The net port area afforded by this valve may be less than in some compressors having a group of inlet valves, but is found to be sufficient, because the inlet is concentrated in a single opening. It is probable that during admission there is less difference between the pressure of the air taken into the cylinder and the atmospheric pressure than with any form of spring-controlled valve; for, meeting with

no resistance due to springs, the air enters freely. Moreover, when the end of the stroke is reached, the inflow of air is momentarily checked, while the piston is reversing, and the momentum of the column of air in the inlet tube tends to cause a slight increase in the density, and therefore in the weight, of the body of air already in the cylinder.

These valves wear well, and their use permits a piston speed high enough to conform with modern high-speed compressor practice. Other advantages are: the cylinder castings are simplified; the space in each cylinder head that would otherwise be occupied by inlet valves is utilized for additional water-jacket area; and the number of moving and wearing parts is reduced. It has been objected that, since the hollow rod and piston are necessarily heated, these advantages are partly offset by a rise in the temperature of the intake air, in its passage into the cylinder; and that therefore the weight of air in the cylinder is relatively less than if it had entered by a more direct path. But it does not appear that this objection is well founded. The air enters the hollow piston rod in a single large volume, instead of being divided into comparatively small areas of flow. It has little opportunity to absorb heat until it reaches the valve, because only a thin film of the rapidly moving air in the inlet tube is in contact with the tube itself.

No positively conclusive tests have yet been made, as to the relative heating of the intake air in "Hurricane-Inlet" and in poppet-valve compressors. Some heat is undoubtedly absorbed by the air in passing in the thin sheet through the valve port in the piston face; but thermometric observations, taken inside the inlet tube and piston, at speeds of 40 and 120 revolutions per minute, show an increase of temperature of not over 5° Fah. at the lower speed and even less at the higher speed. It seems unlikely that any better results are obtainable from either poppet or Corliss inlet valves.

The "Hurricane-Inlet" is a modification of the well-known Ingersoll-Sergeant "Piston-Inlet" Valve, employed for many years in the compressors of these builders. The older type is

somewhat heavier in proportion to the area of opening and is quite different in design. Like the "Hurricane-Inlet," it rests in the seat without springs or other connections; but in the piston casting there is inserted a series of small studs, which pass through slots in the ring valve and so limit the throw.

Johnson Valve. In the Johnson compressor, built in England, there is a single poppet inlet valve of the gridiron type at each

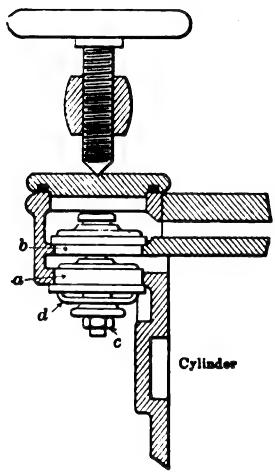


Fig. 65.

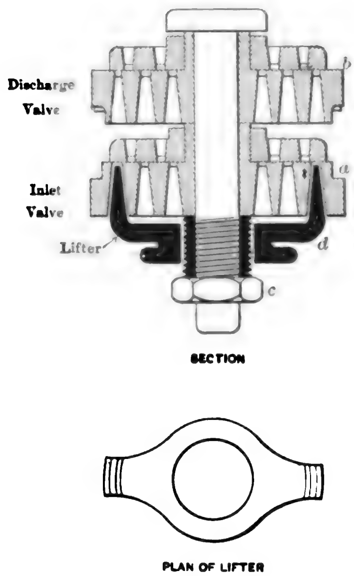


Fig. 66.

Figs. 65 and 66.—Johnson Air Valves.

end of the cylinder. It has a large area, with a small lift, and is mounted in a peculiar way on the same spindle with the discharge valve (Figs. 65 and 66). Both valves are rendered easily accessible by being placed in a chamber projecting horizontally from the end of the cylinder. This chamber is closed by a cast-iron plate held in place by a yoke and set-screw. The lift of the valve is controlled by an outside adjusting nut, *c*, on the spindle. The inlet valve is provided with a "lifter" (Fig. 65, *d*) by which it can be raised trom its seat and thrown out of use, if it be desired tempora-

9

rily to make the compressor single-acting. The Johnson valve closes by gravity only, no springs being used.

Humboldt Rubber Ring Valve. The older form of Humboldt wet compressor (see Fig. 44) has a simple and ingenious valve (Fig. 67). It consists merely of a rubber ring of round cross-section which covers a series of horizontal slots, or ports, in a cylindrical casting set in the top of each air chamber. Three of these rings, *a*, with the slots, *f*, comprise the inlet valves in each end of the air cylinder; the casting, *c*, in which they are placed forming a part of the valve-chamber cover. The casting, *c*, is strengthened

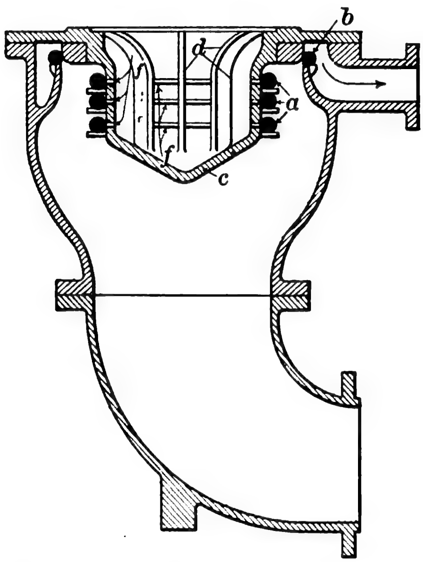


FIG. 67.—Humboldt Rubber Ring Valve.

against the internal pressure by a series of webs, *d*. As the pressure in the air chamber falls during the inlet stroke the atmospheric pressure expands the rubber rings, forcing them away from the slots, and allowing air to enter. Then on the reversal of the stroke the elasticity of the rings causes them to tighten up on their seats and close the ports. The valve openings are relatively large and permit free entrance of air. The discharge valve, *b*, has

the same construction, but consists of a single ring only, of larger diameter and cross-section. These rubber valves are found to last well, as they are kept wet and are not exposed to any great degree of heat. They would be entirely unsuitable for dry compressors. Similar rubber valves are used in a wet compressor built by the Dingler Machine Works, Zweibruecken, Germany.

The Guttermuth Valve is used in a later form of compressor built by the Humboldt Machine Works. It is a spring clack-valve, made of a rectangular plate of thin steel and provided with a grid seat. One side of the plate is coiled in a spiral, through the center of which passes a stationary rod or spindle, the inner edge of the spiral being inserted in a longitudinal groove in this spindle. By placing several valves side by side any desired area of opening can be furnished. To avoid the harmful effects of inertia, the valves are made of extremely thin plate, with delicately adjusted and sensitive springs, and by so arranging them that the current of air in passing through the valve into the cylinder undergoes but slight changes of direction, any serious eddying of the air around the edges of the plate is prevented.

Leyner Flat Annular Valve. This recent form of valve, together with its arrangement on the cylinder heads, is shown by Figs. 68 and 69. Fig. 68 comprises a longitudinal section through the adjacent ends of the low- and high-pressure cylinders of a straight-line, two-stage compressor, indicating incidentally the circulation of the air through the intercooling tubes of both cylinders, as described in Chapter VI. At each end of the cut, left and right, is an outline cross-section, respectively of the low-pressure and high-pressure cylinder heads, showing the groups of intercooler tubes, with the valves themselves and their ports.

The inlet and discharge valves being similar in form, a description of the inlet only will be given (Fig. 69). It consists of a thin steel plate cut in a peculiar form. The outer, or seating portion, is a narrow annulus, with two slender internal arc-shaped strips terminating in a central ring, which is locked against the cylinder head by a steel nut encircling the piston-rod, thus holding the valve in place. The arc-shaped strips, connecting the seating part

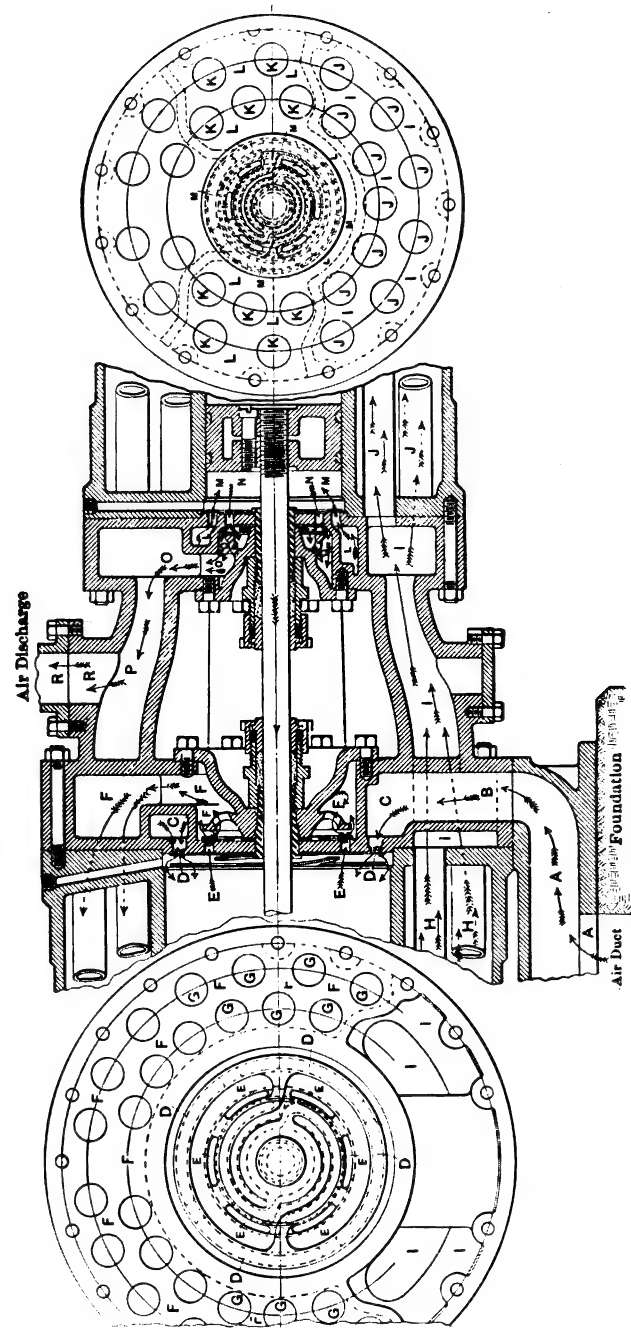


FIG. 68.—Leyner Compressor. Part Section Showing Flat Annular Inlet and Delivery Valves.

of the valve with the fixed central ring, are sufficiently long and flexible to serve as springs, and to permit the valve to open and close freely under very small differences of pressure. There is but one inlet and one delivery valve at each end of the cylinder. The inlet ports, D, D (Fig. 68), four in number in each cylinder head, are curved, slot-like openings, arranged in the form of a circle. There are six similar but smaller discharge ports, E, E. Total area of inlet ports is about fourteen per cent., and of discharge ports, nearly nine per cent. of the piston area. The discharge valves are held in position by the hollow conical casting surrounding the

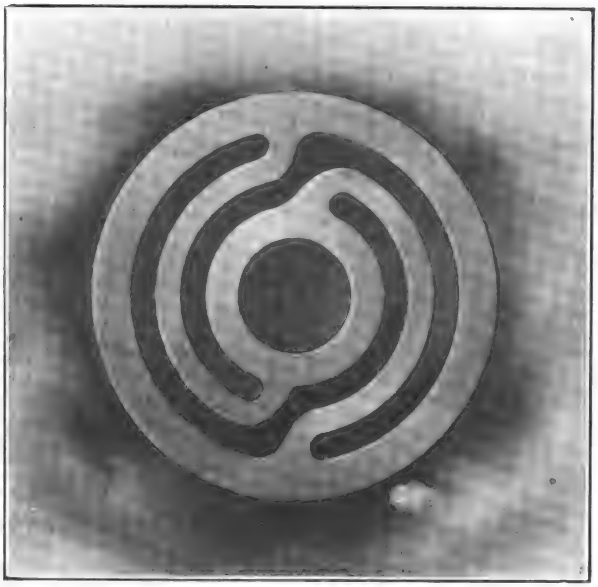


FIG. 69.—Leyner Annular Inlet-Valve

piston-rod stuffing-box. Their height of lift is limited by the stops, shown near F, F.

This valve is simple in design, without separate springs, and consists of one part only. It cannot be doubted that the resistance to opening of the inlet valves is extremely small. As the clearance

volume in these compressors is small (1.02 per cent. of the cylinder volume in the 14-inch high-pressure cylinder mentioned in the foot-note*), a high volumetric efficiency is stated to have been obtained, a number of tests showing it to range from 94.6 to 97 per cent.

Arrangements for Admitting Inlet Air to the Compressor. It is of great importance that the intake air shall be as cool as possible. The colder the air the smaller is the volume occupied by a given weight of air taken into the compressor cylinder, and the greater the output. Taking in warm air involves loss of capacity and of economy in production. Mr. Frank Richards points this out in a convincing and simple way.† "The volume of air at common temperatures varies directly as the absolute temperature. With the air supply at 60° its absolute temperature is 521°, and its volume will increase or decrease $\frac{1}{521}$ for each degree of rise or fall of temperature. Therefore, if in securing the supply of air we can get a difference in our favor of 5° . . . we accomplish a saving of about one per cent. If a difference of temperature of 10° can be secured two per cent. is saved," practically without cost. The practice of taking air from the engine-room is a common one at mines, and is bad not only because such air is usually heated to a considerable degree, but is apt also to be charged with dust which causes unnecessary wear of valves and piston.

Some means should be provided to convey to the compressor fresh air, taken preferably from some point outside of the building. A box or pipe of wood is better than one of iron, because of the smaller conductivity of wood. Its cross-section should be sufficient, say, at least one-half the area of the cylinder, to avoid loss from friction. To make such a connection conveniently the inlet valves should be enclosed in an external air chest on each end of the cylinder. Compressors having a single inlet valve, such as

* In a communication to the author the makers state that repeated tests of a 14 and 22 × 26 inch, 2-stage compressor show a loss of intake pressure of only 0.9 ounce. On a card made with a 20-scale spring, this would be represented by a difference of the inappreciable amount of 0.003 inch, between the intake and atmospheric lines. The frictional loss through the delivery ports of the same compressor is 3 ounces.

† "Compressed Air," p. 55.

the Norwalk, Ingersoll-Sergeant, Sturgeon, etc., are better adapted than some of the others for making this arrangement. In any case, care should be taken to prevent the entrance of dust, leaves, or rubbish. If the inlet be left open, particles floating in the air may be drawn in by the strong current, and obstruct the valves or injure their seats and the smooth working surfaces of piston and cylinder. In such a design as that of the Ingersoll-Sergeant piston inlet, it is essential that the outer end of the hollow rod be covered, because in case of derangement the valves are not so accessible as ordinary poppets. This protection is provided in recent designs of this compressor. By building a suitable conduit from the outside of the compressor house to the air box enclosing the inlet valves, it is obvious that a greater saving can be effected in winter than in summer, but even in warm weather some advantage is gained, especially if the conduit opens on the north side of the building, out of reach of the sun's direct rays, and is carried vertically to some height above the ground level.

CHAPTER VIII

DISCHARGE OR DELIVERY VALVES

THE conditions affecting the action of the discharge valves of a compressor are wholly different from those which govern the suction or inlet valves. While the latter must be capable of opening under very small differences of pressure, the discharge valves are subjected to a heavy pressure on both sides. Furthermore, owing to unavoidable irregularities in the use of the air, the receiver pressure usually fluctuates considerably, so that the point of the stroke at which the discharge valves open cannot depend solely on the conditions, as to the ratio of compression, etc., under which the compressor itself is working. The time of opening must depend also on the relation between the variable pressures in cylinder and receiver.

For this reason, the sensitiveness of operation essential in inlet valves is unnecessary for the discharge valves. The chief requirements are that they shall be free to open when the cylinder pressure exceeds that of the receiver, shall fit accurately on their seats, and close promptly at the end of the stroke. Delay in closing, or leakage between valve and seat, are far more serious than in the case of inlet valves, because these defects are equivalent to an increase of the piston clearance and consequent reduction of the volumetric capacity of the cylinder. The leakage of even a small quantity of compressed air back into the cylinder is equivalent to the loss caused by an abnormally large clearance space. The conditions under which discharge valves operate, therefore, are such as to afford a relatively limited field for innovation or improvement, as compared with inlet valves.

Poppet Discharge Valves. Aside from a few designs in which

mechanical control in some form is introduced (see Chapter IX), nearly all discharge valves are of the poppet type. They are made heavier than inlet valves, with stronger springs to reduce hammering on their seats. Though varying in details of construction, they

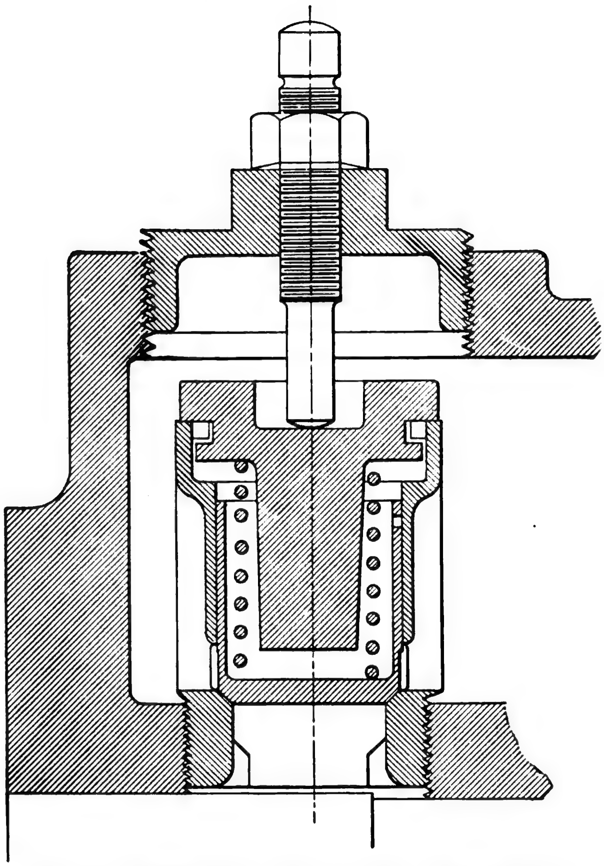


FIG. 70.—Laidlaw-Dunn-Gordon Poppet Discharge Valve.

may be represented fairly by the accompanying figures. Several other designs are also shown in the various sections of air cylinders illustrated in the preceding pages. Two of the ordinary forms of cup-shaped poppet, with internal springs, are shown in Figs. 70 and 71. Occasionally they are of the mushroom type, somewhat

similar in shape to the inlet valve (Fig. 58), the spring then encircling the spindle. The valve may be of steel or bronze, with a bronze seat. To make it easier to keep them tight, the seating surfaces are usually coned. A group of several poppet valves are commonly employed, in order to avoid making them of large size and weight. The inertia of heavy valves causes destructive wear, under their high working pressure. Each valve should be readily accessible for adjustment, re-grinding, or renewal. They are there-

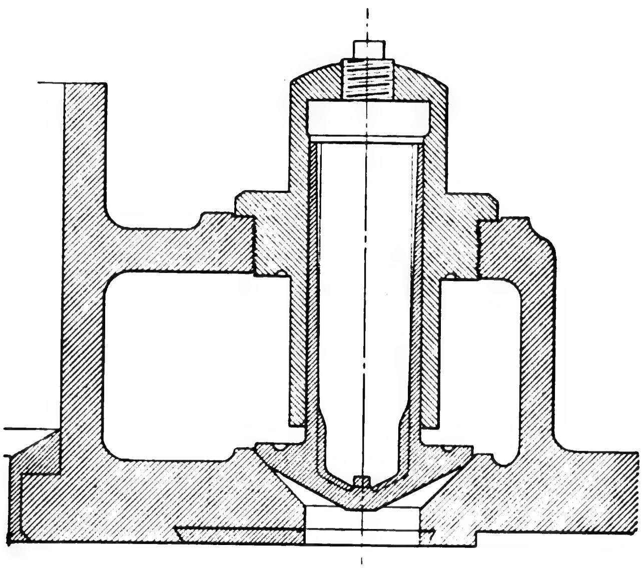


FIG. 71.—Norwalk Poppet Discharge Valve.

fore covered by caps screwed into the outer cylinder head; or, in some makes, by plates bolted on over the valve chamber.

Cataract-Controlled Poppets. In another type of poppet discharge, the valve is not only provided with a spring, but its action is further modified by attaching the valve stem to the piston of a small cataract cylinder, containing either air or oil. This is to ease their movements and avoid hurtful shocks.* Oil-cataract

* Similarly controlled poppets are also employed as inlet valves by some European compressor-builders.

valves are used, for example, in the compressors built by Schuechtermann and Kremer, Dortmund, Germany;* air-cataracts in those of R. Meyer, Muhlheim-Ruhr; G. A. Schuetz, Wurzen; Menck and Hambrock, Altona, and the Humboldt Machine Works, Kalk. (The rubber ring discharge valve, of the last-named builders, has already been referred to, in connection with Fig. 67.)

These valves are employed to a considerable extent in Europe, but are not well known in this country. Some of them are

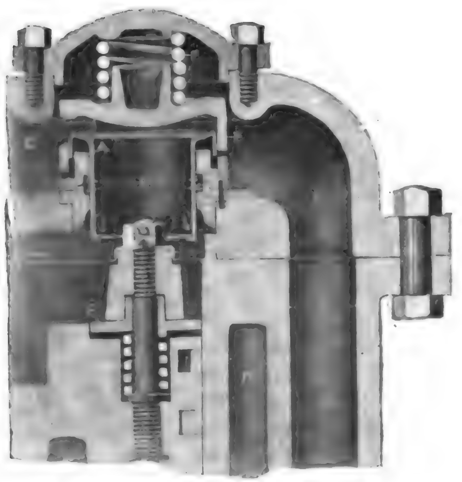


FIG. 72.—"Express" Poppet Valve, Riedler Compressor.

very satisfactory, provided the piston speed be slow; for high-speed compressors they do not work with sufficient promptness to prevent "slip" or leakage of some of the compressed air back into the cylinder. The chief object sought in these cataract movements is attained in another way—by the partial control of an accompanying Corliss valve—in the "Cincinnati" valve gear of the Laidlaw-Dunn-Gordon Co., described in Chapter IX (see Fig. 76).

Riedler Discharge Valve. A poppet discharge valve entirely different in design is shown in Fig. 72, representing one of several patterns employed in the Riedler compressors. This is a light,

* Described in London *Engineering*, Dec. 12th, 1902.

cylindrical valve, A, provided with packing rings D. The cylinder in this case is vertical, and the piston, L, carries at its periphery the plate P, held in place by the stud N and the spring M. When closed the valve seats on the plate E, being held against it by the air pressure in the discharge passage acting on the *under* side of the upper flared end. In this position the round air ports near the lower edge of the valve are closed by the valve guide, at C,C. As the piston advances, and when the cylinder pressure exceeds that in the receiver, the valve is opened by the air pressure on the *upper* side of the flared end. This movement of the valve is cushioned by the air trapped above the guide, B, B. On reaching the end of the stroke, the plate P, on the piston, strikes the lower edge of the valve and closes it against its seat E, the shock being cushioned by the springs F and M. The double cushioning, in both opening and closing, tends to durability; and, moreover, it should be remembered that, when the plate P strikes the valve, the crank is nearly on its center, so that the piston is moving very slowly. The standard mechanically controlled air-valve motion of the Riedler design is described in Chapter IX.

Several other forms of discharge valve will be noted later, in connection with mechanically controlled valve motions.

Discharge Area for Air Cylinders. The volume of air to be discharged from the cylinder having been reduced by compression to a small fraction of the volume occupied at atmospheric pressure, it might appear that the total area of the discharge valves could be made much smaller than the inlet area, without producing excessive frictional resistance. But the compressed air must be forced out of the cylinder in a relatively short period of time. While the air enters throughout nearly the entire stroke, the delivery must take place while the piston is making, say, the last third or quarter of the stroke. Therefore, in a compressor of ordinary design, with several poppet inlet and discharge valves, the total discharge area should be about equal to the inlet area, provided the piston speed be moderate. When the inlet area is concentrated in a single valve (for example, like that of the Ingersoll-Sergeant piston inlet), the discharge area is made about double the inlet area, though this

relation varies in cylinders of different sizes, being proportionately greater in the larger compressors. Obviously, other things being equal, the discharge area should increase with the piston speed. For a speed of 300 feet per minute, the best results are obtained by making the discharge area, say, 10 per cent. of the cylinder area; for speeds of 450 to 500 feet per minute, 15 per cent.* In some compressors, however, the discharge area is as small as from 8.5 to 9.5 per cent.

The above considerations apply in a measure also to the passages through which the air passes from the discharge valves to the pipe leading to the receiver. In some designs these are too restricted to permit a free flow of the air. The velocity of discharge should be made as small as possible, to minimize the resistance due to friction; otherwise, during the period of delivery the pressure of the compressed air in the cylinder will rise momentarily above the normal, and then drop back after the air has passed out to the receiver. This causes a loss of power and unnecessary strains on the moving parts of the compressor. The amount of loss from this cause is represented by the irregular area of the air card which lies above a horizontal line drawn through the point corresponding to the pressure at the end of delivery (see Fig. 61). When the discharge valves first open, the piston is moving at a high velocity, and equilibrium with the receiver pressure is only attained as this velocity decreases toward the end of the stroke.

* W. L. Saunders, "Compressed Air," Dec., 1896, p. 153.

CHAPTER IX

MECHANICALLY CONTROLLED VALVES AND VALVE MOTIONS

THE disadvantageous features of inlet valves whose opening and closure depend primarily upon difference of air pressure have led to the introduction of numerous mechanically controlled valves. By their use fewer valves are required, as a rule, because they may be made much larger and have a higher lift. As distinguished from ordinary poppet valves, they are operated or controlled by being in some way connected with the rotary or reciprocating parts of the compressor. A prompter opening is thus secured, so that the compressor is enabled to take more nearly a full cylinder of air at each stroke.

In some designs the connection between the valves and their operating mechanism is absolutely positive and fixed for any one setting of the valves, which are timed with respect to the piston stroke, so as to open at the instant the clearance air has been re-expanded to atmospheric pressure, and to close at the end of the stroke. Other designs involve the use of springs, which modify to some extent the operation of the controlling mechanism, thus allowing for variations in working conditions, as well as for inaccuracies of adjustment or slight derangements caused by wear of parts. Still other valve motions exert a partial control, which, within narrow limits, leaves the valve free to act under difference of air pressure inside and outside of the air cylinder.

As a rule, in the recent designs of mechanical valve motions the inlet valves only are positively controlled, and in most cases the type of valve used is a modified form of the Corliss. But while mechanically controlled valves are often employed for the low-

pressure cylinders of stage compressors, they are not suitable for the high-pressure cylinders; the inlets of these are subjected to heavy pressures on both sides, and are best allowed to open and close solely under the difference between these pressures, which is more than sufficient to produce prompt action of the valve at the proper time. Poppet valves are therefore generally used for this service.

Mechanical Control for Discharge Valves. The adoption of any system of mechanical control for discharge valves is a matter of some difficulty, because of the fluctuations of receiver pressure under which these valves are compelled to operate. In attempting to open them by a positive mechanical movement, at a fixed point of the stroke, two cases may occur: 1, in event of a drop in receiver pressure below the normal, the valves and their controlling mechanism would be subjected to a heavy strain, before the point of opening is reached, due to the excess of cylinder pressure; and, 2, if the pressure in the receiver should rise above the normal, the valves would be held forcibly on their seats, by the excess of receiver pressure, after being released by the controlling gear. In either case, derangement or breakage of the valves or of some part of the controlling mechanism may occur.

Hence, in order that the discharge valves may adjust themselves automatically to the varying conditions, some degree of freedom as to their time of opening must be allowed. It is true that the range of fluctuation in receiver pressure is lessened by the use of air-pressure regulators (Chapter XII); nevertheless, only a partial mechanical control of these valves is practicable for any service in which the consumption of air is variable or intermittent. Moreover, Corliss valves of the ordinary patterns used for compressors do not serve well for discharge valves where the ratio of compression is greater than, say, three or three and one-half; because they must then be set to open too late in the stroke to permit a free discharge. This applies to single-stage compressors, as designed for ordinary service, as well as to the high-pressure cylinders of two-stage machines. A number of devices have been introduced for dealing with these conditions; such as the use of relief valves working in conjunction with mechanically operated discharge

valves; or, as in one form of the Riedler compressor, the opening of the valve is governed in part by the air pressure, a very small free lift being allowed by the controlling mechanism for affording the necessary relief.

Valve Motion of Norwalk Compressor. An adaptation of the Corliss valve gear has been used for many years for the low-pressure cylinder of this compressor (Fig. 73). One large inlet and one discharge valve are set in chests at each end of the cylinder. Poppet valves are employed for the high-pressure cylinder. These are shown in Fig. 8, together with the cross-sections of the low-pressure Corliss valves in their repective chests. The main valve-rod, *a*, Fig. 73, is driven by a drag- or return-crank, *b*, mounted on the crank-pin of the fly-wheel. The rod is pin-connected to a short lever, *c*, on the spindle of the forward inlet valve, and from this lever a link, *d*, passes to a corresponding connection with the inlet valve at the other end of the cylinder, the parts being so adjusted that one valve opens as the other closes. A positive movement of the valves is thus obtained.

An essential feature of this valve motion is the introduction of the cams *f f* and *g g*, for operating the discharge valves. These cams are mounted in pairs on the respective inlet and discharge valve spindles, and form part of the short levers *c*. As each inlet valve oscillates, its cam rolls smoothly upon that of the discharge valve above it, the shape of each pair of cams being such that the discharge valve is opened full at the proper point of the stroke, *i.e.*, when the pressure within the cylinder becomes equal to that in the discharge passage outside. Then, at the end of the stroke, when the cams move in the opposite direction, and while still rolling upon each other, the discharge valve is closed without shock by the connecting link, *e*. This link is elastic, being made of two telescoping parts, somewhat on the principle of a dash-pot, thus allowing the freedom of movement necessary for dealing with variable receiver pressure.

In recent years, a number of other compressor-builders have adopted modifications of the Corliss valve gear for the air cylinders.

Nordberg Valve Motion. For single-stage compressors of this

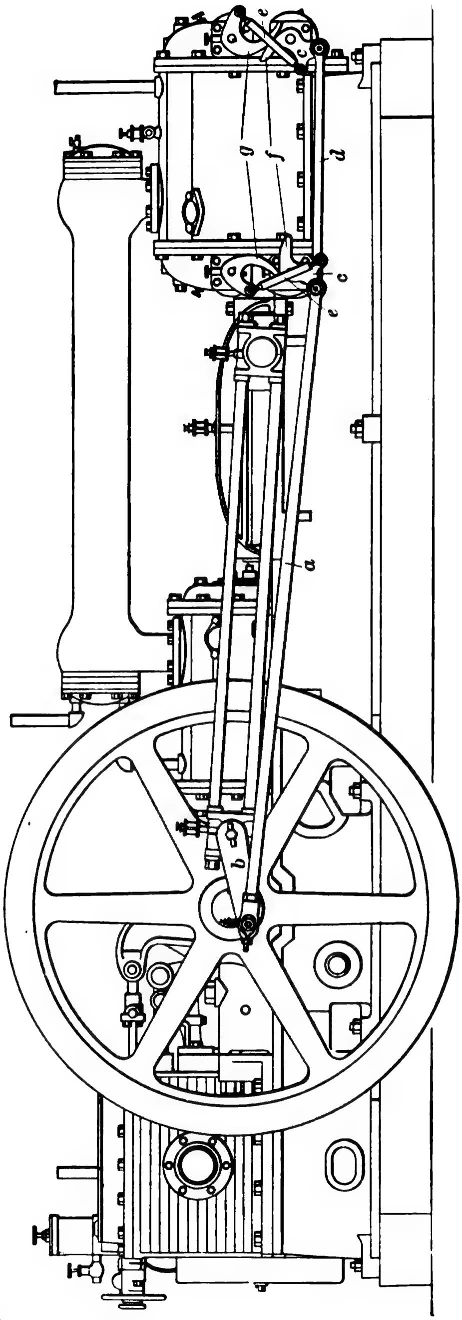


FIG. 73.—Valve Motion of Low-Pressure Air Cylinder. Norwalk Compressor.

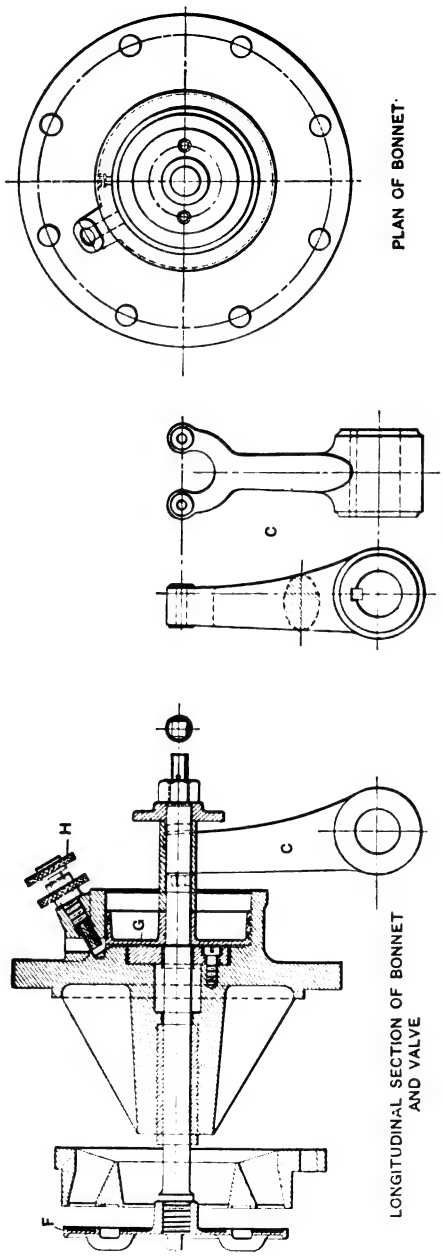


Fig. 80.—Details of Riedler Inlet Valve.

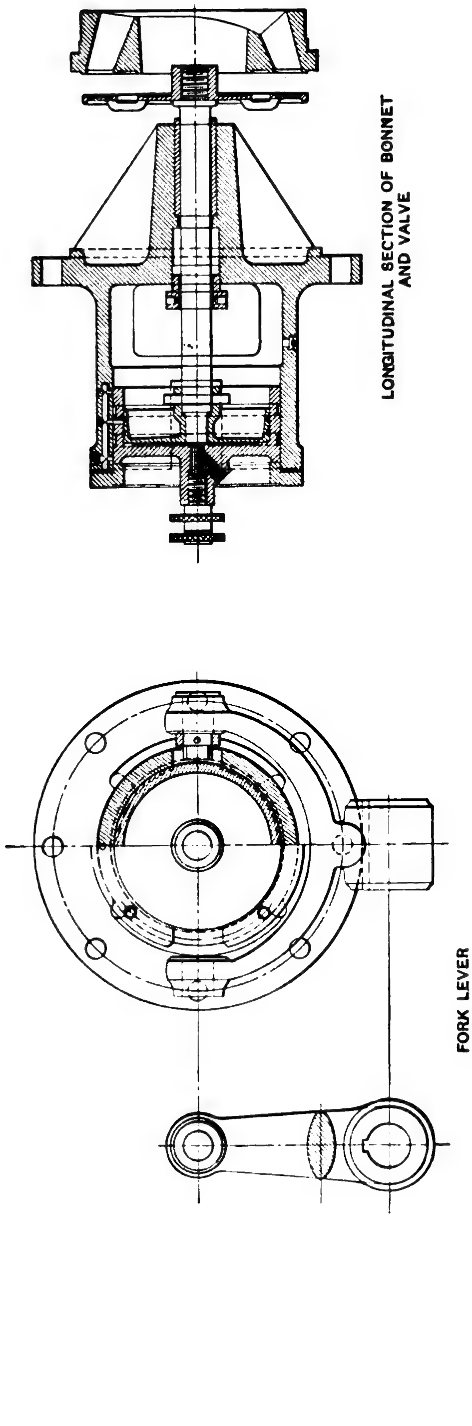


Fig. 81.—Details of Riedler Discharge Valve.

open, its movement being steadied by the dash-pot piston, G. The resistance presented by this piston is regulated by the adjusting screw, H. For ordinary sizes of compressor the total lift of the valve is one inch, giving a large area of opening. (The 10¼-in. valve shown in the cut, which is for the low-pressure cylinder of a 24″ and 38″ × 48″ compressor, has an area of 45 sq. ins.) Toward the end of the stroke the forked lever begins to rise, thereby bringing the valve gradually nearer its seat, as the piston velocity decreases. In completing its movement the lever forces the valve upon its seat promptly at the end of the stroke. By this device, the valve attains its maximum lift and area of opening toward the middle of the stroke, when the velocity of the inflowing air is greatest, and is brought nearer its seat as the flow diminishes, so that the complete closure is effected instantaneously at the proper time.

A similar control is exerted over the delivery valve, though the details of its bonnet, dash-pot, and forked lever are quite different, as shown by Fig. 81.* At the proper point of the stroke the lever is depressed, so that the valve is free to open when the air pressure in front of the advancing piston has reached receiver pressure. Then, as the velocity of outflow diminishes toward the end of the stroke, the valve is forced nearer its seat, and a prompt closure takes place the instant the stroke reverses. As a result of this mechanical control, together with the action of the dash-pot, the operation of the Riedler valves is attended with but little shock, thus permitting a high piston speed.

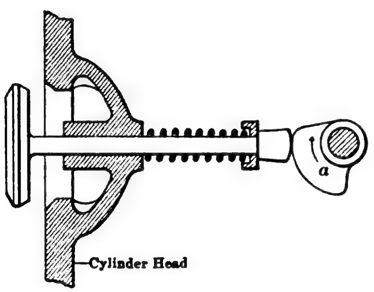


Fig. 82.—Cam-Controlled Inlet Valve.

Cam-Controlled Inlet Valve. At the Lens colliery, in France, a cam movement has been successfully applied for controlling the opening of a poppet inlet valve (Fig. 82). The stem of the valve

* The delivery valve in the cut is the same size as the suction valve previously described. It is designed for a smaller compressor, 15″ and 24″ × 36″.

is provided with a spiral spring, and projects from the cylinder head, as usual. At the beginning of the stroke the valve is opened rapidly by a cam, *a*, of peculiar shape, playing against the end of the stem. The cam is mounted upon a small shaft which is geared to revolve once for each revolution of the compressor. At the end of the stroke the cam allows the valve to close under the action of the spring.*

Sturgeon Inlet Valve. This peculiar valve, of an air compressor made in England, furnishes an example of a positive movement entirely different in principle from those already described. It is a large annular valve, *c* (Fig. 83), encircling the piston rod in each cylinder head, and is operated directly by the movements of the rod itself.† The connection between the valve and rod is frictional only, being brought about by a gland, *e*, which serves also to form a stuffing-box for the piston. By tightening or loosening the nuts, *a*, of the bolts by which the gland is attached to the valve flange, any desired amount of grip upon the piston rod can be obtained. This frictional grip is regulated so that the valve will not be opened until the clearance air has been re-expanded nearly to atmospheric pressure. Flanges on the ends of the valve limit its play in each direction, controlling the amount of lift and area of opening. The valve and stuffing-box together form the bearing of the piston rod in each cylinder head. At the end of the stroke a recess in the piston receives the large inner flange of the valve so as to diminish the clearance. The mechanism is simple and its work satisfactory.

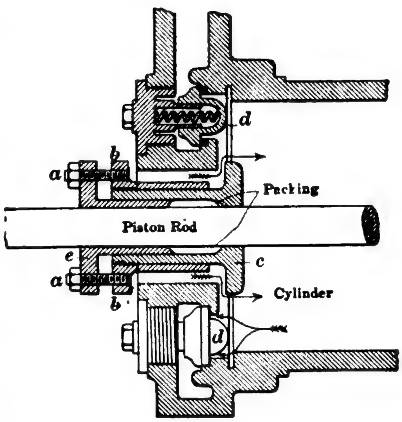


FIG. 83.—Sturgeon Inlet Valve.

* H. W. Hughes, "Text-book of Coal Mining," p. 55. † *Idem*, p. 53.

An arrangement resembling the above has been used in a compressor made by the Dover Iron Co., Dover, N. J.

The well-designed Köster valve is of the piston type, almost unknown in this country for air-compressor service. It is now employed by several European makers, among them: Pokorny & Wittekind, Frankfort, Neumann & Esser, Aachen, and W. H. Bailey & Co., Manchester, England. The valves, both inlet and discharge, are very large in area and are mounted on a longitudinal spindle deriving its reciprocating motion from an eccentric on the crank-shaft. Positive opening and closure are imparted to the inlet valves, but the opening of the delivery port is effected by an independent poppet, encircling the spindle and provided with a light spring. This valve motion constitutes a highly developed type, and is both reliable and efficient.

CHAPTER X

PERFORMANCE OF AIR COMPRESSORS*

THE performance or duty of air compressors may be designated in several different ways.

First. A standard of rating, useful for ordinary purposes, is the duty in terms of cubic feet of free air compressed per minute to a given pressure; or, the volumetric output. The theoretical output is found by multiplying the net piston area in square feet by the distance travelled by the piston in feet per minute. The actual output will be less than the theoretical on account of various losses due to leaks, clearance, induction of warm air, friction of inlet valves, etc. In a properly designed compressor an allowance of fifteen per cent. to eighteen per cent. is sufficient to cover these losses, which must not be confounded with the mechanical loss of work—that is, the work expended in overcoming the friction of the compressor—and the loss of *useful* work due to the heating of the air under compression.

Having found the capacity of the compressor, in terms of cubic feet of free air, the volume, V', occupied by this air at any given pressure, P', is calculated by the formula already given: $V' = \frac{V P}{P'}$, in which the following values are now assigned, *viz.*:

V = initial volume of given quantity of air in cubic feet.

P = normal absolute pressure of atmosphere (14.7 lbs.).

P' = absolute pressure of air under compression, *i.e.*, gauge pressure + 14.7 lbs.

For example, 100 cu. ft. of free air, compressed isothermally to 65 lbs. gauge pressure, will occupy a volume:

$$V' = \frac{100 \times 14.7}{65 + 14.7} = 18.45 \text{ cu. ft.}$$

* The deductions of the work formulæ used in this chapter are given in detail in Chapter III.

Conversely, the volume of free air represented by 18.45 cu. ft. of air at 65 lbs. gauge pressure is:

$$V = \frac{V'P'}{P} = \frac{18.45\ (65 + 14.7)}{14.7} = 100 \text{ cu. ft.}$$

By applying the 15 to 18 per cent. allowance for losses stated above, this allowance depending on the type of compressor, results are obtained sufficiently accurate for practical purposes. As the volumetric output of a compressor of given size of cylinder depends on the density of the intake air, it will obviously be reduced when working at an altitude above sea-level. (See Chapter XIII.)

Second. The size of the compressor may be designated in terms of the horse-power developed by the steam end, indicator cards being taken while running at normal working speed and while the usual volume of air is being consumed.

Third. The effective horse-power of the quantity of compressed air delivered is determined from an indicator card, taken from the air cylinder. In testing a compressor it is customary to take a series of cards, simultaneously from both ends of the steam and air cylinders. They may then be compared, as shown by Fig. 24.

If indicator cards be not available, the theoretical horse-power for single-stage adiabatic compression may be calculated by the formula:

$$\text{H. P.} = \frac{144\ P\ V\ n}{33{,}000(n-1)} \left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right], \text{ in which}$$

P = normal atmospheric pressure per square inch (14.7 pounds).

P' = final absolute pressure per square inch.

V = the volume of free air compressed per minute, in cubic feet.

n = exponent of the compression curve, as given under the theory of air compression, *viz.:* for adiabatic compression, $n = 1.406$, and varies down to 1.18 or 1.2, depending on the efficiency of the cooling arrangements. For the best single-stage compressors, n = say, 1.25 or 1.3.

For isothermal compression, the expression for the horse-power is:

$$\text{H.P.} = \frac{144}{33{,}000} \times P\ V \left(\text{Nap. log. } \frac{P'}{P}\right)^{*}$$

Table V shows the horse-powers required, under the conditions named, to compress one cubic foot of free air per minute to different gauge pressures, by single-stage compression:

TABLE V

Gauge Pressure, Lbs.	Atmospheres Absolute, or Ratio of Compression.	Single-Stage Compression, from Atmospheric Pressure at Sea-Level. Initial Temp., 60° Fah. Horse-Power Required to Compress 1 cu. ft. of Free Air.			
		Theoretical Horse-Power.		Actual Horse-Power (Approx.).	
		Isothermal Compression.	Adiabatic Compression.	Allowance for Losses above Adiabatic Compression, 15%.	Allowance for Losses above Adiabatic Compression, 20%.
20	2.36	.0551	.0626	.0720	.0751
25	2.71	.0637	.0741	.0852	.0890
30	3.04	.0713	.0843	.0970	.1011
35	3.38	.0782	.0941	.1082	.1129
40	3.72	.0842	.1029	.1183	.1234
45	4.06	.0895	.1115	.1282	.1338
50	4.40	.0950	.1191	.1370	.1430
55	4.74	.0994	.1269	.1460	.1522
60	5.08	.1041	.1337	.1537	.1604
65	5.42	.1081	.1401	.1610	.1681
70	5.76	.1123	.1468	.1690	.1761
75	6.10	.1162	.1535	.1765	.1842
80	6.44	.1195	.1591	.1830	.1910
85	6.78	.1224	.1651	.1900	.1961
90	7.12	.1256	.1703	.1955	.2040
95	7.46	.1287	.1760	.2024	.2112
100	7.80	.1315	.1807	.2080	.2168
110	8.48	.1366	.1894	.2180	.2272
125	9.50	.1442	.2025	.2328	.2430

In columns three and four of Table V are shown the theoretical horse-powers required for isothermal and adiabatic compression. The results of isothermal compression are wholly unattainable in practice, and are placed here only for purposes of comparison. They represent an ideal which it is desirable always to keep in view.

* The Naperian or hyperbolic logarithm of a number is obtained by multiplying the common logarithm by the constant 2.302585.

The figures given in the column of adiabatic compression are based on the assumptions that there is no radiation of heat from the air cylinder, and that the temperature of the air after delivery has become normal, its volume being therefore reduced to that which is practically available for use. No allowances are included in these figures to cover losses other than that due to the heating of the air under compression. But the full amount of loss represented by adiabatic compression can never be suffered in the operation of compressors, however imperfect their design. The actual compression line must always be lower than the adiabatic line, because of the radiation of heat through the cylinder walls. In ordinary, single-stage compressors, properly water-jacketed and run at a reasonable piston speed, the compression line falls considerably below the adiabatic line. Whatever diminution of loss is effected by cooling of the air in the cylinder may therefore be credited against the other unavoidable losses, partially offsetting them, *viz.:* frictional or mechanical loss in the compressor, friction of inlet valves, heating of the intake air by contact with the hot metal surfaces, and piston clearance of the cylinder. These losses are variable in amount, depending on the design of the compressor.

In the absence of indicator cards, giving the actual results in individual cases, estimates based on practice may be made of the net power loss experienced in operating compressors, which will be convenient for reference. With this understanding, an attempt is made, in columns five and six of the above table, to show the actual horse-power required to compress one cubic foot of free air, under the conditions stated at top of the columns. Thus, in column five, fifteen per cent. is assumed as a fair estimate, in case of well-designed and operated single-stage compressors, of the additional power required, over and above that for theoretical adiabatic compression; this fifteen per cent. being taken as: the loss in purely adiabatic compression, minus the effect of ordinary water-jacket cooling, plus the other four losses mentioned at end of preceding paragraph. In column six, the power consumed in adiabatic compression is increased by twenty per cent., which represents relatively poorer work.

The figures in columns 3 and 4 or 5 and 6 (which are for *free* air), if multiplied by the corresponding ratios of compression (column 2), will give the respective theoretical and actual power costs of furnishing one cubic foot of *compressed* air, at the gauge pressures stated.

Work Done by Stage Compressors. The theoretical horse-power required to compress a given volume of free air to any given pressure, P′, is computed for a two-stage compressor by the formula:

$$\text{H.-P.} = \frac{2 \times 144}{33{,}000} \times \frac{PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{2n}} - 1\right]$$

This formula is derived from that for single-stage compression by dividing equally between the cylinders the total work done, and then taking the sum of the two.

For three-stage compression the formula becomes:

$$\text{H.-P.} = \frac{3 \times 144}{33{,}000} \times \frac{PVn}{n-1}\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{3n}} - 1\right]$$

Reducing the constants, and for a volume of one cubic foot of free air, these formulas may be simplified thus:

$$\text{Two-stage, H.-P.} = 0.449\left[\left(\frac{P'}{P}\right)^{0.144} - 1\right]$$

$$\text{Three-stage, H.-P.} = 0.6735\left[\left(\frac{P'}{P}\right)^{0.062} - 1\right]$$

In these expressions it is assumed that the work of compression in each cylinder is done adiabatically, and that the temperature of the air after leaving the cylinder is reduced by intercooling to the initial temperature.

For convenience, the horse-powers for stage-compression at sea-level, both theoretical and actual, are given for a few gauge pressures in the following table; the figures in the fifth and seventh columns being taken as an approximation to the results obtainable in practice from stage compressors of the usual designs.

At elevations above sea-level, P is less than 14.7, and for any given altitude the atmospheric pressure must therefore be known.

TABLE VI

Gauge Pressure, Lbs.	Ratio of Compression $=\frac{P'}{P}$	Horse-Power per Cubic Foot of Free Air.				
			Two-Stage Compression.		Three-Stage Compression.	
		Isothermal Compression.	Adiabatic Compression.	Actual H.-P., on basis of Adia. Comp'n + 18%.	Adiabatic Compression.	Actual H.-P., on basis of Adia. Comp'n + 15%.
70	5.76	0.1123	0.129	0.152		
80	6.44	.1195	.138	.163		
90	7.12	.1256	.147	.173		
100	7.80	.1315	.154	.182	0.145	0.167
120	9.16	.1420	.169	.199	.158	.182
140	10.50	.1508	.181	.213	.169	.194
160	11.88	.1583	.192	.226	.179	.206
180	13.24	.1654	.202	.238	.188	.216
200	14.60	.1720	.212	.250	.196	.225
250	18.00	.1853	.232	.274	.213	.245
300	21.40	.1963	.249	.294	.228	.262
350	24.80	.2058	.264	.312	.241	.277
400	28.20	.2140	.277	.327	.252	.290
450	31.62	.2215	.289	.341	.262	.301
500	35.01	.2280			.271	.311
550	38.41	.2339			.280	.222
600	41.80	.2393			.288	.331
650	45.21	.2443			.295	.339
700	48.62	.2490			.301	.346
800	55.42	.2574			.314	.361

Table VII will be found useful for making calculations in which are used volumes and mean cylinder pressures for isothermal and adiabatic compression.

In this table the mean pressures per stroke, given in the fifth and sixth columns, are obtained from the formulas for isothermal and adiabatic single-stage compression, which precede Table V, except that they are here expressed in terms of foot-pounds of work, instead of horse-power. These formulas may be put respectively in the following forms:

$$\text{Mean pressure per stroke (isothermal)} = P \times \text{Nap. log. } \frac{P'}{P}$$

$$\text{Mean pressure per stroke (adiabatic)} = 3.463\, P \left[\left(\frac{P'}{P}\right)^{0.29} - 1 \right]^{*}$$

* The constant $3.463 = \frac{n}{n-1} = \frac{1.406}{.406}$

TABLE VII *

Gauge Pressures.	Atmospheres.	Volume with Air at Constant Temperature.	Volume with Air Not Cooled.	Mean Pressure per Stroke; Air at Constant Temperature. Pounds.	Mean Pressure per Stroke; Air Not Cooled. Pounds.	Temperature of Air; Not Cooled. Degrees Fahrenheit.
0	1	1	1	0	0	60°
1	1.068	.9363	.9500	.96	.975	71
2	1.136	.8803	.9100	1.87	1.91	80.4
3	1.204	.8305	.8760	2.72	2.80	88.9
4	1.272	.7861	.8400	3.53	3.67	98
5	1.340	.7462	.8100	4.30	4.50	106
10	1.680	.5952	.6900	7.62	8.27	145
15	2.020	.4950	.6060	10.33	11.51	178
20	2.360	.4237	.5430	12.62	14.40	207
25	2.700	.3703	.4940	14.59	17.01	234
30	3.040	.3289	.4538	16.34	19.40	252
35	3.381	.2957	.4200	17.92	21.60	281
40	3.721	.2687	.3930	19.32	23.66	302
45	4.061	.2462	.3700	20.57	25.59	321
50	4.401	.2272	.3500	21.69	27.39	339
55	4.741	.2109	.3310	22.76	29.11	357
60	5.081	.1968	.3144	23.78	30.75	375
65	5.423	.1844	.3010	24.75	32.32	389
70	5.762	.1735	.2880	25.67	33.83	405
75	6.102	.1639	.2760	26.55	35.27	420
80	6.442	.1552	.2670	27.38	36.64	432
85	6.782	.1474	.2566	28.16	37.94	447
90	7.122	.1404	.2480	28.89	39.18	459
95	7.462	.1340	.2400	29.57	40.40	472
100	7.802	.1281	.2320	30.21	41.60	485
105	8.142	.1228	.2254	30.81	42.78	496
110	8.483	.1178	.2189	31.39	43.91	507
115	8.823	.1133	.2129	31.98	44.98	518
120	9.163	.1091	.2073	32.54	46.04	529
125	9.503	.1052	.2020	33.07	47.06	540
130	9.843	.1015	.1969	33.57	48.10	550
135	10.183	.0981	.1922	34.05	49.10	560
140	10.523	.0950	.1878	34.57	50.02	570
145	10.864	.0921	.1837	35.09	51.00	580
150	11.204	.0892	.1796	35.48	51.89	589
160	11.880	.0841	.1722	36.29	53.65	607
170	12.560	.0796	.1657	37.20	55.39	624
180	13.240	.0755	.1595	37.96	57.01	640
190	13.920	.0718	.1540	38.68	58.57	657
200	14.600	.0685	.1490	39.42	60.14	672

* Kents' "Mechanical Engineers' Pocket Book." Taken from a table in Richards' "Compressed Air," p. 20.

The work done during one stroke of the compressor is found by multiplying the mean pressure by the volume in cubic feet, V, traversed by the piston.

When air is compressed adiabatically, the relation between the temperature T, of the air at the beginning of compression, and the temperature at the end, T′, is shown by:

$$\frac{T'}{T} = \left(\frac{V}{V'}\right)^{n-1}, \text{ whence } T' = T\left(\frac{V}{V'}\right)^{n-1}$$

The final temperature may also be found from the formula:

$$T' = T\left(\frac{P'}{P}\right)^{\frac{n-1}{n}}$$

T and T′ being absolute temperatures and P, P′ absolute pressures in each case.

The compression curve of an air-indicator card may be constructed as follows, P V being the pressure and volume at one point of the curve and P′ V′ the pressure and volume corresponding to any other point. Designating the index number of the curve by x:

$$\frac{P}{P'} = \left(\frac{V'}{V}\right)^{x}. \quad \text{From this,}$$

$$\log.\left(\frac{P}{P'}\right) = x \log.\left(\frac{V'}{V}\right); \text{ whence, } x = \frac{\log.\left(\frac{P}{P'}\right)}{\log.\left(\frac{V'}{V}\right)}$$

In considering an air card, it should be observed that the several lines have significations entirely different from those of a steam card. Referring to Fig. 84, which represents an ideal card: A B is the admission line, B C the compression line, C D the delivery or discharge line, and D A the re-expansion line. The last-named line represents the effect of the re-expansion of the air filling the clearance space in the cylinder, on beginning a stroke (see Chapter V). Comparing the lines of the air and steam cards, they are found to be reversed, thus:

Air Card.	Steam Card.
Admission line.	Back-pressure or exhaust line.
Compression line.	Expansion line.
Delivery line.	Admission line.
Re-expansion line.	Compression line.

The elements of an air-indicator card, together with the work done, as represented by the several lines and areas, will be further elucidated by referring to Fig. 85.

In this analysis the compression is supposed to be done adiabatically.

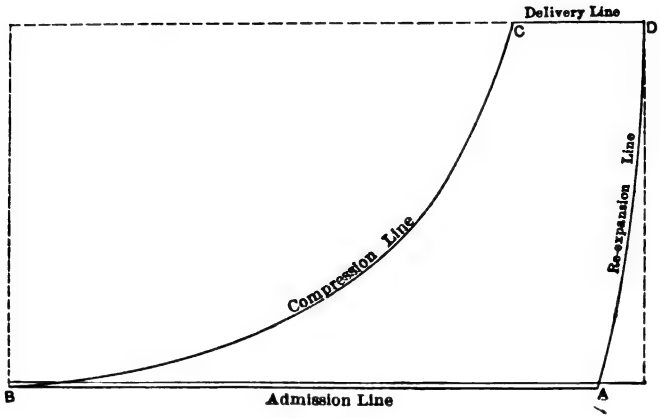


FIG. 84.

Let A D = normal atmospheric line at sea-level.

A G = P = corresponding atmospheric pressure, acting behind the piston at the beginning of the stroke (neglecting valve resistances and effect of clearance of previous stroke).

G E = A D = length of stroke of piston.

A B = adiabatic compression curve.

B C = delivery line.

At the point B the useful work of compression ceases; during the remainder of the stroke the volume of compressed air V', at the absolute pressure P', is being forced out of the cylinder through the delivery valves.

The area A B F G = the absolute work of compression.

The area B C E F = the absolute work of delivery.

The sum of these areas represents the total absolute work (that is, on the basis of absolute pressure) done during compression and delivery.

Area A D E G = work done for the entire stroke by atmospheric pressure behind the piston.

Area A B H = net work of compression.

Area B C D H = net work of delivery.

Area A B C D A = total net work for entire stroke.

From this analysis another method may be derived for calculating the theoretical horse-power required for compressing air.

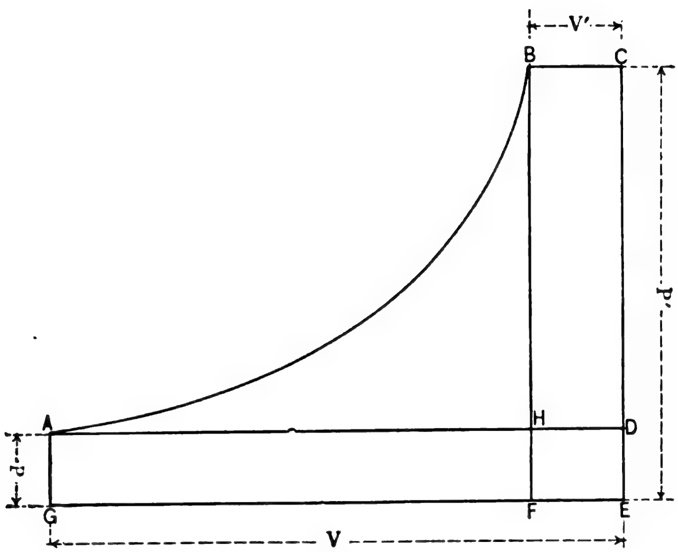


FIG. 85.

It will often be found useful, when a table of temperatures of compression is available.

Let w = weight of a unit volume (1 cu. ft.) of free air = .0765 lbs.

C_p = specific heat of air at constant pressure = .2375.

C_v = specific heat of air at constant volume = .1689.

$$\text{Whence, } \frac{C_p}{C_v} = n = 1.406, \text{ and } \frac{n}{n-1} = 3.46.$$

J = Joule's heat unit, taken as 778 ft. lbs. The work represented by the area A B H =

$$J \times w \times C_v (T' - T) - P (V - V').$$

Also, the work done during delivery = B C D H = $V'(P' - P)$. Hence, the total net work for one stroke of the piston

$$= \text{area } A\,B\,C\,D\,A = J \times w \times C_v (T' - T) - (P\,V - P'\,V').$$

If C_p be substituted for C_v, then $P\,V = P'\,V'$, according to the general equation for air compression and the total work, W=

$$J \times w \times C_p (T' - T).$$

Substituting for J, w, and C_p, their constant numerical values:

$$W = 14.13\,(T' - T),$$

or, to compress 1 cu. ft. of air per min., at 60° F., and at sea-level,

$$\text{H.-P.} = 0.225 \left[\frac{T'}{T} - 1\right]$$

By referring to the last column of Table VII and remembering that T and T′ are absolute temperatures, *i.e.*, thermometric temperatures plus 459° F., the horse-power required for compressing one cubic foot of free air adiabatically to any gauge pressure may readily be calculated.

Other expressions, for the mean effective pressures, may also be deduced from what precedes:

M.E.P. for the entire stroke =

$$P\frac{n}{n-1}\left(\frac{T'}{T} - 1\right) = 3.46\,P\left(\frac{T'}{T} - 1\right) = 3.46\,P\left[\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} - 1\right]$$

$$\text{M.E.P. during delivery} = \frac{V'}{V}(P' - P).$$

The M.E.P. for compression only is found by taking the difference between the pressures calculated by the last two formulas.

The results obtained from the above expressions for work and mean effective pressure are theoretical. To find the actual horsepower required, allowances must be made for the several losses experienced in the operation of the compressor, as already set forth.

Compresssor Tests. To indicate the observations required to secure the data for the complete test of a compressor, together with the deductions from the observed data, the following record of the test of a compound, two-stage Nordberg compressor, at

the mines of the Tennessee Copper Co., will be found useful.* It will be noted that items 28, 29, and 32 to 35, are necessary in this case, because the boiler plant supplied steam for the hoisting engine and an independent condenser, as well as for the compressor. Though the hoist was not running, steam was passing continuously to the jackets of the cylinders. The same conditions would often be met in other tests. The boiler feed water was taken from a wooden tank, and during the run this water was supplied from two barrels on scales set temporarily over the tank. The water of condensation from steam jackets and reheater was drawn off continuously and also weighed. The calorimeter tests were made with a Peabody throttling calorimeter. Eight sets of indicator cards were taken during the 8-hour test, at hourly intervals.

Items of Compressor Test

Altitude, 1,800 feet.

1.	Date of test, February 16, 1902	
2.	Duration of test, hours	8
3.	Diameter of high-pressure steam cylinder (steam jacketed), inches	14
4.	Diameter of low-pressure steam cylinder (steam jacketed) inches	28
5.	Diameter of low pressure air cylinder, inches	24½
6.	Diameter of high pressure air cylinder, inches	15⅜
7.	Stroke of all pistons, inches	42
8.	Diameter of piston rods, inches	2$\frac{11}{16}$
9.	Revolutions of engine, average per minute	90
10.	Piston speed per minute, feet	630
11.	Steam-gauge pressure, average, pounds	145.9
12.	Temperature of steam in steam pipe, average degrees Fah.	364
13.	Steam pressure in reheating receiver, average pounds	8
14.	Vacuum in condenser, average inches	25.66
15.	Air pressure in intercooler, average, pounds	22.63
16.	Air pressure in receiver, average, pounds	79.3
17.	Temperature of air at intake, average degrees Fah	65.0
18.	Temperature of air leaving low-pressure cylinder, average degrees Fah	211.5
19.	Temperature of air leaving intercooler, average, degrees Fah	78.5
20.	Temperature of air leaving high-pressure cylinder, average, degrees Fah	240.0

* Abstracted from an article by J. Parke Channing, *Mines and Minerals*, May, 1905, p. 475.

21.	Indicated horse-power in high-pressure steam cylinder, average	140.12
22.	Indicated horse-power in low-pressure steam cylinder, average	153.03
23.	Indicated horse-power in both steam cylinders, average	293.15
24.	Indicated horse-power in low-pressure air cylinder, average	143.79
25.	Indicated horse-power in high-pressure air cylinder, average	135.02
26.	Indicated horse-power in both air cylinders, average	278.81
27.	Feed-water weighed to boilers, pounds	43,343
28.	Re-heater and jacket water from compressor, weighed, pounds	4,081
29.	Average temperature of re-heater and jacket water, degrees Fah.	356.7
30.	Total heat in 1 pound of steam at 356.7 degrees Fah., heat units	1,190.7
31.	Total heat in 1 pound of water at 356.7 degrees Fah., heat units	328.9
32.	Equivalent credit for re-heater and jacket water, pounds	1,127.00
33.	Water weighed from condensation in hoisting-engine jacket, pounds	1,781.00
34.	Steam used to run condenser, pounds	4,320.00
35.	Total credits to feedwater, pounds	7,228.00
36.	Total feedwater charged to engine, pounds	36,115.00
37.	Moisture in steam shown by Peabody calorimeter, per cent	1.30
38.	Credit for moisture in steam, pounds	473.00
39.	Total steam charged to engine, pounds	35,642.00
40.	Dry steam per hour charged to engine, pounds	4,455.00
41.	Steam consumption per indicated horse-power per hour, pounds	15.19
42.	Guaranteed steam consumption per indicated horse-power per hour, at 92 revolutions per minute, pounds	14.00
43.	Excess of steam consumption per indicated horse-power per hour over guarantee, pounds	1.19
44.	Theoretical delivery of free air per minute at 90 revolutions, cubic feet	2,037.8
45.	Slip of air (percentage)	3.0
46.	Actual slip of air per minute, cubic feet	61.1
47.	Actual delivery of free air per minute, average cubic feet	1,976.7
48.	Theoretical horse-power required to compress and deliver actual delivery of air at receiver pressure by adiabatic compression	306.53
49.	Theoretical horse-power required to compress and deliver actual delivery of air at receiver pressure by isothermal compression	229.00
50.	Actual horse-power shown by air indicator cards	278.81
51.	Actual horse-power shown by steam indicator cards	293.15
52.	Actual horse-power consumed by friction of engine	14.34
53.	Efficiency ratio between steam and air cylinders, per cent	95.1
54.	Efficiency ratio between steam and air cylinders guaranteed by builder, per cent	87.0
55.	Efficiency of steam, or ratio of steam indicated horse-power to theoretical air indicated horse-power, isothermal compression, per cent	78.1

One of the combined indicator cards, from which the averages in items 21 to 26 were calculated, is shown in Fig. 86.

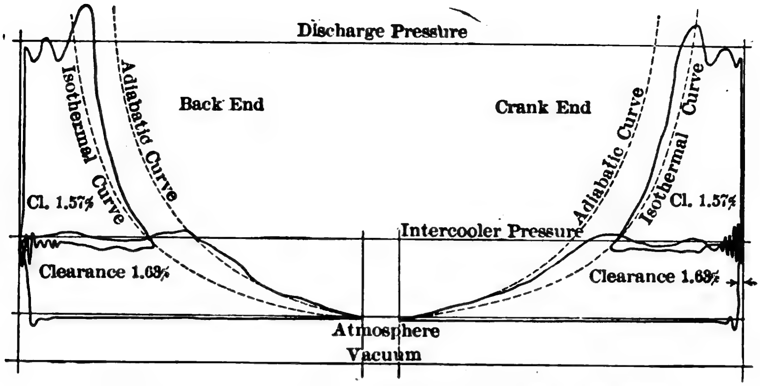


FIG. 86.—Combined Cards. Two-Stage Nordberg Compressor.

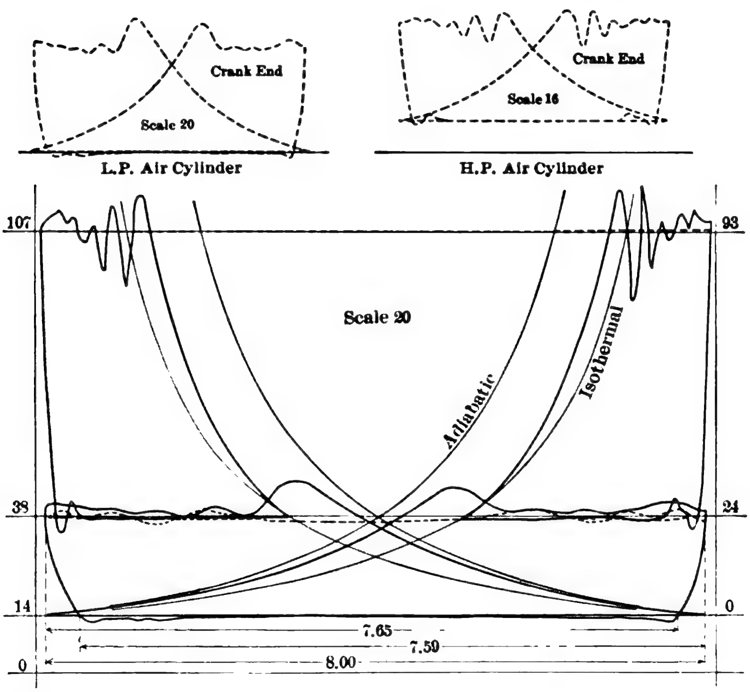


FIG. 87.—Combined Cards, "Imperial Type 10," Two-Stage, Direct-Connected, Electrically Driven Compressor. Air Cylinders 23" and 14" × 20".

Revolutions per minute	187.	I.H.P. of high-pressure cylinder	120.8
Piston speed, feet per minute	623.3	Total I.H.P.	252.8
Discharge air pressure, lbs	93.	Free air delivered per minute cubic feet (from card)	1706.
Intercooler pressure, lbs	24.	Efficiency compared with adiabatic	97.2%
Volumetric efficiency (from card)	95.3%	Efficiency compared with isothermal	84%
I.H.P. of low-pressure cylinder	132.		

In further illustration of the performance of air compressors, the combined card from an Ingersoll-Rand "Imperial" two-stage compressor, taken at one of the Berwind-White Coal and Coke Company's mines, is given in Fig. 87.

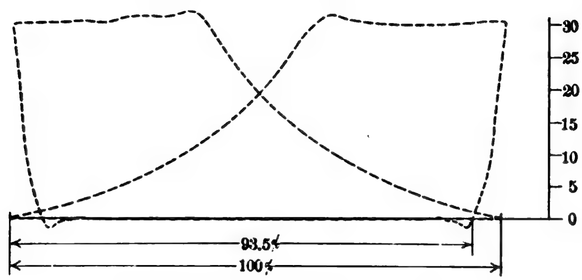


FIG. 88.—Card from $30\frac{1}{4}$"× 24" L. P. Air Cylinder of Style "O," Ingersoll-Rand Compressor. (St. press., 115 lbs.; air press., 28 lbs.; r. p. m., 100; spring, 20.)

Figs. 88 and 89 show shop-test cards, with accompanying data, from the high- and low-pressure air cylinders of an Ingersoll-Rand style "O" compressor.

A Record of Field Tests. It would undoubtedly tend to the securing of greater economy in the production of compressed

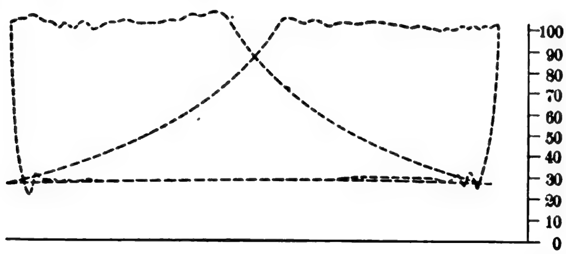


FIG. 89.—Card from $18\frac{1}{4}$" × 24" H. P. Air Cylinder, Style "O," Ingersoll-Rand Compressor. (St. press., 115 lbs.; air press., 100 lbs.; r. p. m., 100; spring, 60.)

air, if superintendents and master mechanics would give more attention to the actual results produced by the operation of compressors in their charge, and study carefully the frequently unfavorable conditions under which these machines are called upon to work.

Few records of the actual effective horse-power of air compressors have been published. To express the efficiency, it is customary to divide the horse-power of the air cylinder by the horse-power of the steam cylinder, as determined by indicator cards. The manufacturer of air compressors usually rates his machine on the basis of its mechanical efficiency, without taking into consideration any losses except those of friction. Such a criterion does not properly measure the relative commercial values of compressors, nor does it present any indication as to the effective horse-power developed under ordinary working conditions.

A series of tests were made in 1909 by Richard L. Webb, consulting engineer, of Buffalo, N. Y., on a large number of compressors in a well-known Canadian mining district. In conducting these tests, Mr. Webb had access to plants which have been in operation for a year or more under normal working conditions, and I believe his results will be of value not only to users of air compressors, but also to the manufacturers. As a rule, the plants tested were in the care of competent machinists and in good running order, so that the results obtained may be taken as representing a fair average of current practice in the United States and Canada. The results of a few of these tests are given here to show the importance of determining the actual efficiency of air compressors when working under the conditions prevailing in most mines.

Mode of Conducting the Tests. The following plan was employed in each case. *First,* a boiler test was run for not less than two weeks, the coal being carefully weighed, the boiler feed water measured, and the total revolutions of the compressor recorded by a revolution counter. From these data, the cost per boiler horse-power and the average speed of the compressor were determined. *Second,* the compressor was operated at various speeds over its entire range. By means of a meter installed in the steam pipe near the throttle, the total steam consumed, in pounds per hour, was measured. Indicator cards were taken on all cylinders, together with temperatures at the air inlet, intercooler, and discharge. To measure the actual volume of air delivered, a meter was placed in

the discharge pipe outside of the receiver. A number of simultaneous readings on all instruments were taken at each speed. From these were calculated the total horse-power of the steam and air cylinders, the steam consumption, and the total piston displacement per minute.

The air and steam meters were of the Dodge type, as modified by the General Electric Company, and were operated by their

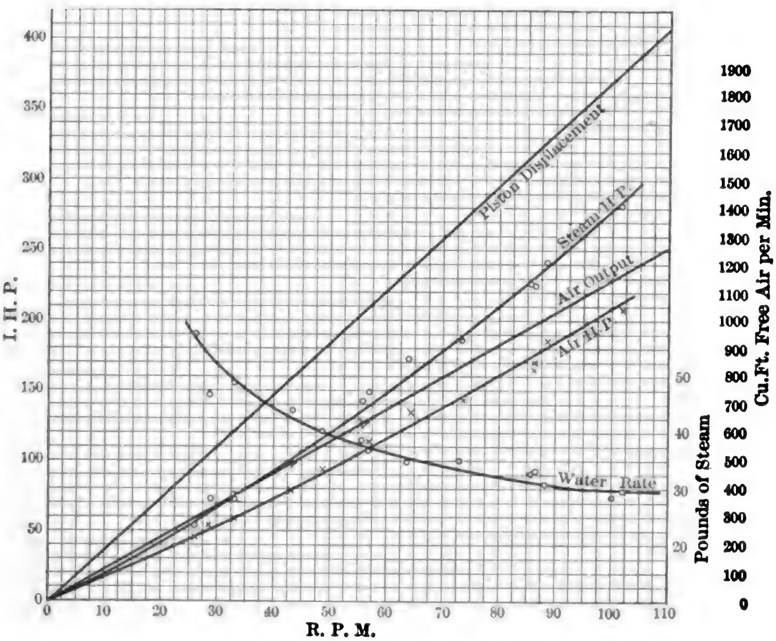


FIG. 90.—Compressor Plant No. 1.

expert sent for this purpose. The indicators were of the Roberts-Thompson and the American-Thompson make, which are well known and generally accepted as standard. Their springs were calibrated by a standard gauge.

Results of the Tests. As was to be expected, the friction loss was found to be only a small item in the total. The other losses, which are frequently overlooked or disregarded, played a large part in cutting down the efficiency. The capacity of air com-

pressors is usually rated according to the volume of the cylinders. On this basis, the mechanical efficiency only is given. For example, if the horse-power of the air cylinder is 100 horse-power and the horse-power of the steam cylinder 110, the efficiency of

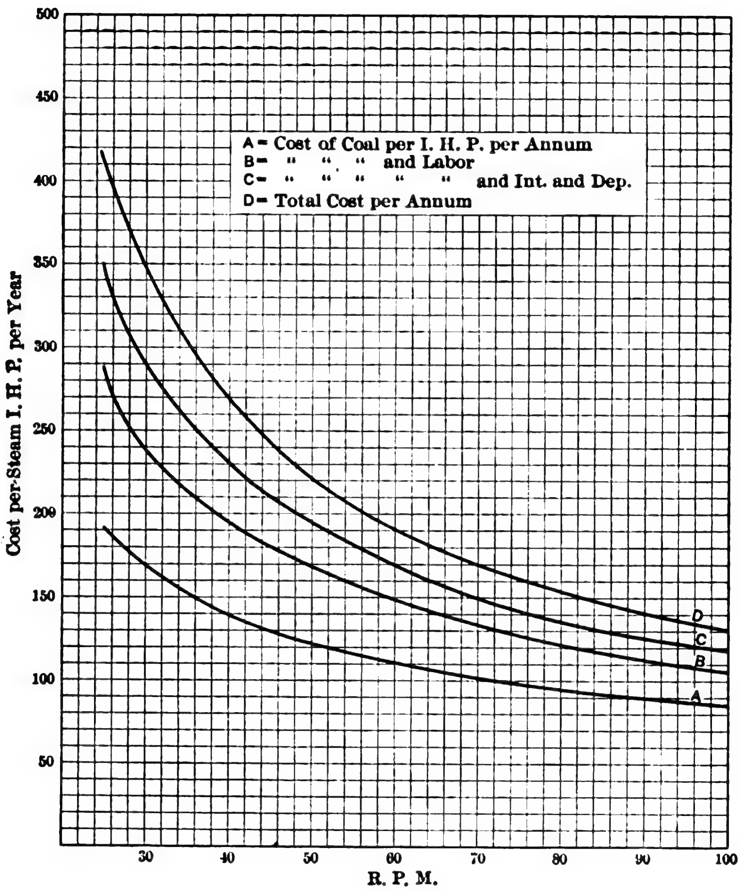


FIG. 91.—Compressor Plant No. 1.

the compressor is rated as 91 per cent. This rating disregards the losses due to adiabatic compression, heating of the cylinder and friction of the inlet and delivery valves. The tests show the friction loss of the engine itself to be usually not less than 10

per cent., and often considerably larger. Losses from the other causes mentioned were found to range from 30 per cent. up.

As Mr. Webb is not at liberty to disclose the identity of the particular plants at which the tests were made, each test has been designated by a number.

Test of Plant Number One. This consists of three 125 H.P. return tubular boilers (one being held in reserve), supplying steam for a cross-compound condensing air compressor of standard make. The steam cylinders have Meyer valve gear and are 16″ and 28″ diameter by 24″ stroke. The two-stage air cylinders are 28″ and 18″ by 24″. From a two weeks' run the following results were obtained.

Total coal burned, lbs	264,300
Total feed water, cu. ft	37,459
Total feed water, lbs	2,335,568
Average temperature of feed water, degrees Fah	131
Average evaporation per lb. coal consumed, lbs	8.72
Average revolutions per minute	63.1
Indicated horse-power of steam end, corresponding to 63.1 R.P.M. steam	161
Corresponding indicated horse-power of air end	123
Average steam pressure, lbs	115
Average vacuum, lbs.	10.5
Average air pressure, lbs	96
Average temperature of outside air, degrees Fah	24
Average air piston displacement at 70° F., cu. ft	1172
Average metered output corrected to 70° F., cu. ft	758

The average evaporation, of 8.72 lbs. of water from 131° F. to an average steam pressure of 115 lbs., is equivalent to 9.83 lbs. of water evaporated from and at 212° F. per lb. coal consumed. At the average compressor speed of 63.1 revolutions per minute, the metered output was equivalent to 758 cubic feet of free air per minute, the piston displacement being 1,172 cu. ft. per minute. Table VIII and Fig. 90 present the principal data of the test run on this compressor.

To find the average operating results, the curves at 63 revolutions should be followed, at which the indicated H.P. of the steam cylinder was 160.8, and that of the air cylinder, 123, showing the mechanical efficiency to have been 76.5 per cent. The theoretical

TABLE VIII.—TEST ON PLANT NO I.

COMPOUND CONDENSING STEAM CYLINDERS, 16" AND 28", WITH MEYER VALVE GEAR; AIR CYLINDERS, 28" AND 18", STROKE 24".

Revolutions per Minute.	Steam Pressure, Pounds.	Air Pressure, Pounds.	High-Pressure Side. Steam. M.E.P.	High-Pressure Side. Steam. I.H.P.	High-Pressure Side. Air. M.E.P.	High-Pressure Side. Air. I.H.P.	Low-Pressure Side. Steam. M.E.P.	Low-Pressure Side. Steam. I.H.P.	Low-Pressure Side. Air. M.E.P.	Low-Pressure Side. Air. I.H.P.	Total Steam. I.H.P.	Total Air. I.H.P.	Friction Loss.	Friction Loss. Per cent.	Air Line, Temperature.	Cu. Ft. Displacement. 70° F.	Metered Air Output per Min. Cu. Ft. 70° F.	Calculated Output. Cu. Ft.
25.8	125	75	55.0	34.	33.95	20.68	10.3	19.75	14.31	21.68	53.75	42.36	11.39	21.2	220	480	340	295
32.7	126	87	63.2	49.5	39.65	30.65	10.7	26.0	14.69	28.2	75.5	58.85	16.65	22.05	245	610	415	375
43.0	126	95	63.5	65.45	42.44	43.10	11.65	37.2	14.16	35.73	102.7	78.8	23.9	23.2	250	800	519	492
55.5	129	100	64.6	86.0	46.15	60.4	13.61	56.2	15.32	49.9	142.2	110.3	31.9	22.4	258	1030	605	635
64.0	128	100	65.65	100.7	48.25	72.8	15.1	71.8	15.85	59.5	172.5	132.3	40.2	23.3	268	1190	650	730
73.0	126	94	61.2	107.1	42.88	73.8	14.78	80.2	15.64	67.0	187.3	140.8	46.5	24.8	262	1355	690	835
88.0	125	97	61.62	129.9	48.4	100.5	17.12	112.0	16.56	86.7	241.9	186.2	55.7	23.0	286	1655	740	898
101.08	127	95	63.85	155.9	47.75	114.7	16.5	124.9	15.8	94.5	280.8	209.2	71.6	25.5	280	1890	760	1165

NOTE—The above data are from a table of fourteen readings taken at various speeds.

TABLE IX.—TEST ON PLANT NO. I.

POWER COSTS. DUPLEX MEYER STEAM, COMPOUND AIR END. OPERATED CONDENSING.

Revolutions per Minute.	Evaporation.	Water Rate.	Pounds Coal per I.H.P. Hour.	Steam I.H.P.	Pounds Coal per Hour.	Coal. Cost per Annum.	Coal. Cost per I.H.P. per Annum.	Labor per I.H.P.	Coal and Labor per I.H.P.	Int. and Depreciation per I.H.P.	Coal, Labor, Int. and Depreciation per I.H.P.	Repairs, etc., per I.H.P.	Total Cost per I.H.P.	Cu. Ft. Compressed Air per Hour.	Operating Hourly Cost.	Cents per 1,000 Cu. Ft.
25	8.08	59	7.30	55	401.5	$10,550	$192.00	$97.00	$289.00	$62.00	$351.00	$62.00	$414.00	2963	2.59	.874
40	8.31	44	5.30	95	503.5	13,240	130.00	56.00	195.00	36.00	231.00	36.00	268.00	4238	2.90	.684
60	8.62	36.5	4.23	152	642.9	16,900	111.00	35.00	146.00	23.00	169.00	23.00	191.00	5963	3.32	.557
80	8.82	32	3.67	211	774.4	20,350	96.45	25.00	121.00	16.00	137.00	16.00	154.00	7650	3.72	.487
100	8.96	29.5	3.29	278	914.6	24,050	86.60	19.00	106.00	12.00	118.00	12.00	130.00	9308	4.14	.445

Table X.—Test on Plant No. 2.

Duplex Simple Steam Cylinders, 14" × 22", Meyer Valve Gear; Two-Stage Air End, 22" and 14" × 22" Stroke.
Rated Capacity, 1,050 Cu. Ft. per Minute, at 105 R.P.M.

Revolutions per Minute.	Gauge Pressure, Pounds.		High-Pressure Side.				Low-Pressure Side.				Steam Total.	Air Total.	Per cent. Air of Steam.	Pounds Steam, I.H.P. Hour.	Metered Air Output, Cu. Ft. Per Minute, 70° F.	Displacement, Cu. Ft. per Minute, 70° F.
			Steam.		Air.		Steam.		Air.							
	Steam.	Air.	M.E.P.	I.H.P.	M.E.P.	I.H.P.	M.E.P.	I.H.P.	M.E.P.	I.H.P.	I.H.P.	I.H.P.				
14.3	90	100	44.62	11.3	43.8	10.7	43.2	10.55	14.4	8.71	21.85	19.4	88.9	78.9	295	151
46.0	90	95	45.5	37.0	41.1	32.4	43.3	34.0	15.75	30.6	71.0	63.0	88.8	46.32	564	484
51.0	85	98	45.4	39.6	41.6	36.3	43.9	38.2	15.06	33.4	77.3	69.7	90.3	53.9	654	535
82.0	85	55	42.2	59.1	25.6	35.9	32.5	45.5	15.8	54.6	104.6	90.5	86.5	42.4	744	864
103.0	77	93	48.9	86.1	40.1	70.5	44.1	78.2	17.03	74.0	164.3	144.5	88.0	43.1	1000	1083
111.0	82	75	46.7	88.6	32.3	61.4	40.5	76.9	16.8	78.5	165.5	139.9	84.6	38.9	1025	1161

Note.—The above data are from a table of readings taken at fourteen different speeds.

H.P. required to compress isothermally one cubic foot of free air per minute to 96 lbs. (the average pressure) is 0.129. The theoretical useful work done by the compressor is, therefore,

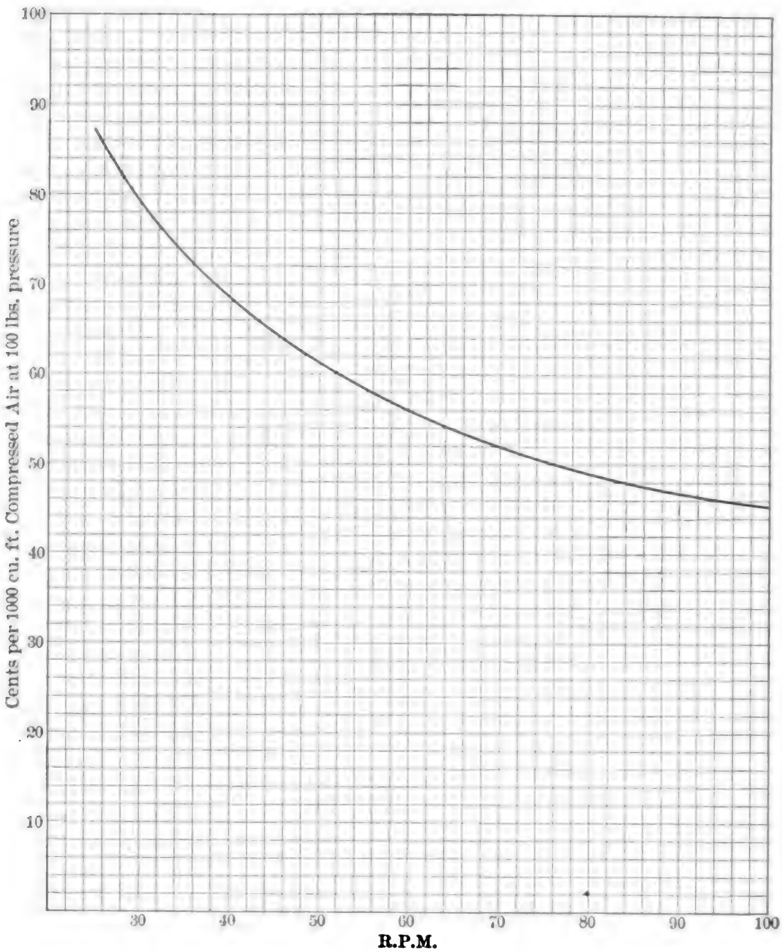


FIG. 92.—Compressor Plant No. 1.

758 × .129 or 97.8, and the net total efficiency of the compressor is 97.8/161 or 60.8 per cent.

Table IX shows the actual cost of running this compressor

at different speeds. The data were furnished by the owner and are based on one year's operation. In Fig. 91 these costs are plotted, showing how the cost per steam H.P. per year is affected

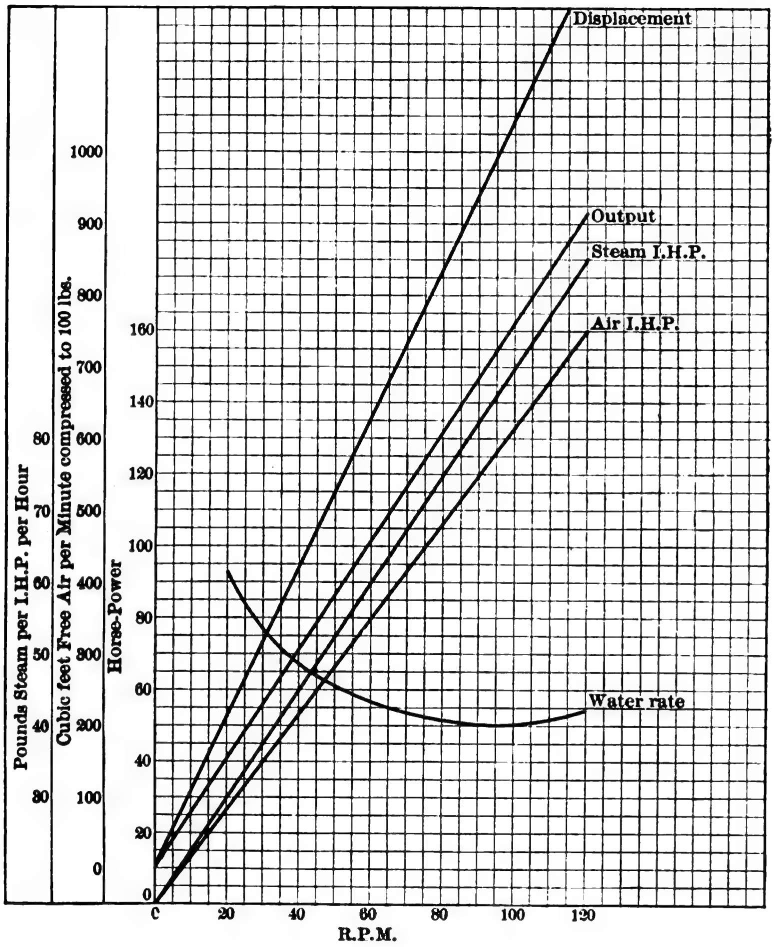


FIG. 93.—Compressor Plant No. 2.

by the average running speed of the compressor. The curve of Fig. 92 shows the operating costs in another way. These costs may be read in terms of 1,000 cu. ft. of free air compressed

to 100 lbs. or 1,000 cu. ft. of compressed air at 100 lbs. gauge pressure.

Test of Plant Number Two. The plant consisted of three 150 horse-power, return tubular boilers, supplying steam for a

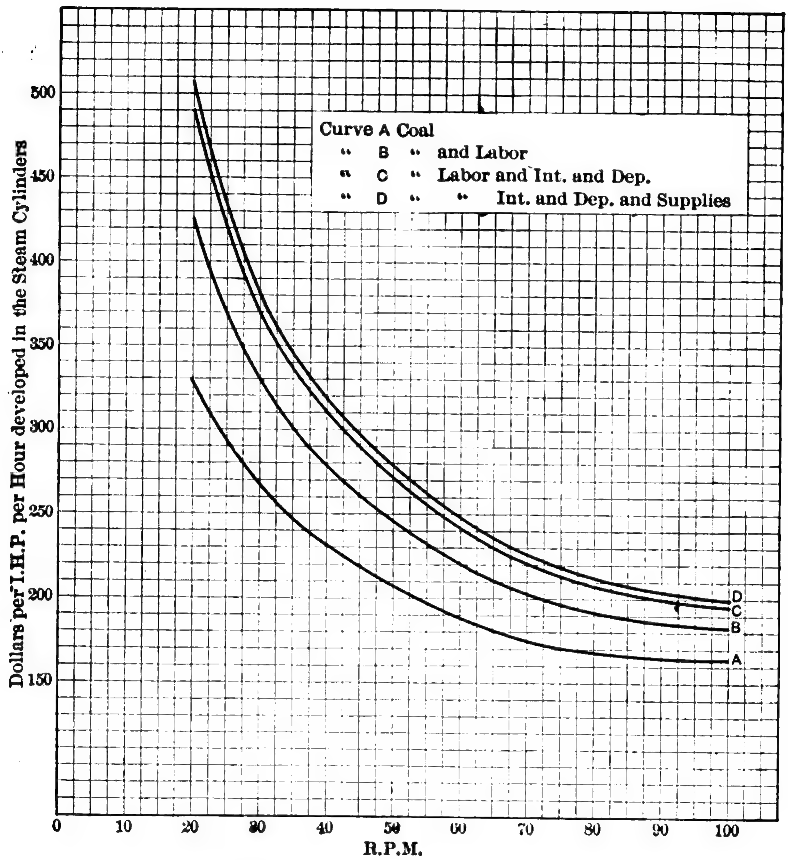


FIG. 94.—Compressor Plant No. 2.

Corliss engine, the air compressor, and steam heating. To determine the boiler horse-power, a meter was placed on the steam pipe to the compressor during the test run, so that only the portion of steam actually used by the compressor was charged to the same. The compressor was duplex, with Meyer valve gear, simple steam

cylinders 14″ × 22″, and two-stage air end, 14″ and 22″ × 22″ stroke, rated by the manufacturer at 1,050 cubic feet of free air per

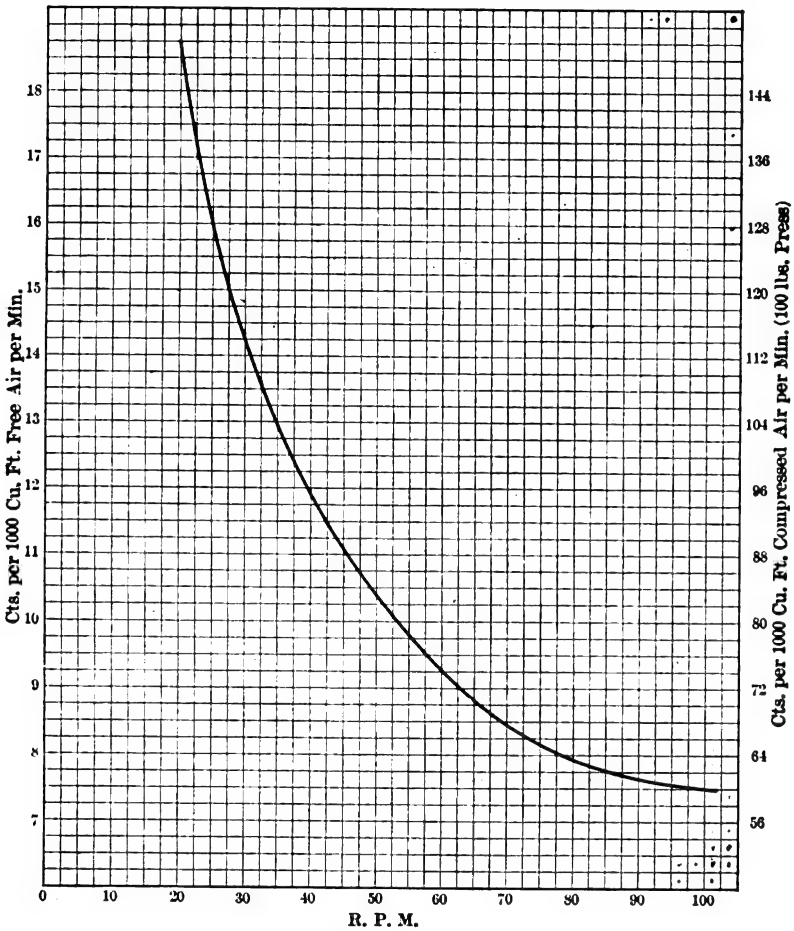


FIG. 95.—Compressor Plant No. 2.

minute at 105 revolutions. At this plant the test lasted over a month, with the following results:

Total coal consumed, lbs.	459,250
Total feed water, lbs.	2,496,000
Average evaporation per lb. coal consumed, lbs.	5.46
Average revolutions per minute	36.
Corresponding average indicated horse-power (from curve)	53.

Hourly readings of the revolution counter were taken, showing an average speed of 36.05 revolutions. At this speed the steam consumption was 51 pounds per I.H.P. hour, as measured at the throttle, the air meter showing a delivery of 275 cubic feet of free air per minute. The total efficiency was 67 per cent. Taking the ordinary method of computing the mechanical efficiency only at the same speed, there would be 48 air horse-power, divided by 54 steam horse-power, giving an efficiency of 89 per cent.

The coal consumption per indicated horse-power per year, as shown by the books of the company, amounted at the average speed to about 56 tons. Table X, with Figs. 93, 94, and 95, present the details of the test on this plant, which was conducted in a manner similar to that on plant No. 1.

Test of Plant Number Three. This plant consisted of two 125 horse-power return tubular boilers, supplying steam for a non-condensing cross-compound air compressor of standard make; steam cylinders 18″ and 35″×24″, air cylinders 14″ and 28″×24″. A two weeks' run gave the following results:

Total coal burned, lbs	221,190
Total feed water, cu. ft.	34,273
Total feed water, lbs	2,094,657
Average temperature feed water, degrees Fah	154
Average evaporation per lb. coal consumed, lbs	9.48
Average boiler horse-power	208
Average revolutions per minute	66
Average indicated horse-power of steam end, at 66 R.P.M (from curve)	210
Average indicated horse-power of air end (from curve)	128.5
Average steam pressure	97
Average air pressure	97
Average outside temperature, degrees Fah	23
Average air piston displacement at normal speed, cu. ft., at 70° F.	1,372
Metered output in cu. ft. corrected to 70° F.	734

The average evaporation of 9.48 lbs. of water per pound of coal, from 154° F. to an average steam pressure of 97 pounds, is equivalent to 10.4 pounds of water evaporated from and at 212° F. At the average speed of 66 revolutions, the displacement was 1,240 cubic feet of free air per minute, while the metered output was 734 cubic feet, showing a net volumetric efficiency of 59 per cent.

TABLE XI.—TEST ON PLANT NO. 3.

COMPOUND NON-CONDENSING STEAM CYLINDERS, 18" AND 35"; TWO-STAGE AIR END, 28" AND 14" BY 24" STROKE.

Revolutions per Minute.	Steam Pressure, Pounds.	Air Pressure, Pounds.	High-Pressure Side.				Low-Pressure Side.				Total Steam, I.H.P.	Total Air, I.H.P.	I.H.P. Friction.	Per Cent. Loss in Friction.	Air Line Temperature.	Cu. Ft. Displacement, 70° F.	Output Cu. Ft. Air per Minute, 70° F.
			Steam.		Air.		Steam.		Air.								
			M.E.P.	I.H.P.	M.E.P.	I.H.P.	M.E.P.	I.H.P.	M.E.P.	I.H.P.							
24	100	73	53.6	38.6	35.7	15.6	8.93	24.8	13.85	24.7	63.4	40.3	23.1	36.4	120	447	269
29.6	98	67	50.55	44.9	28.5	15.4	7.32	25.0	10.43	29.5	69.9	44.9	25.0	35.8	119	550	330
50	97	100	61.6	92.4	45.55	41.5	10.67	61.7	14.48	53.7	154.1	95.2	52.9	38.2	120	935	555
61	100	102	60.6	110.9	45.8	50.9	11.6	81.8	15.1	68.4	192.7	119.3	73.4	38.1	140	1145	677
75	99	100	62.6	140.9	44.68	61.0	11.76	102.0	15.36	85.5	242.9	146.5	96.4	39.7	160	1409	831
94.5	95	99	48.7	138.0	44.95	77.4	16.31	178.3	15.64	109.7	316.3	187.1	129.2	40.85	165	1780	1050
103.5	98	95	47.7	148	43.0	81.1	17.2	206	16.31	125.3	334.0	206.4	147.6	45.92	173	1948	1150

NOTE.—The above data are from a table of eighteen different readings taken at various speeds.

TABLE XII.—TEST ON PLANT NO. 4.

DUPLEX COMPOUND NON-CONDENSING CORLISS CYLINDERS, 18" AND 30"; TWO-STAGE AIR END, 26" AND 16½" BY 30" STROKE.

Revolutions per Minute.	Steam Pressure, Pounds.	Air Pressure, Pounds.	Temperature Air at Receiver.	High-Pressure Side.				Low-Pressure Side.				Total Steam, I.H.P.	Total Air, I.H.P.	Loss in Friction. I.H.P.	Friction Loss, Per Cent.	Displacement, Cu. Ft.	Output Cu. Ft. Air per Minute, 70° F.
				Steam.		Air.		Steam.		Air.							
				M.E.P.	I.H.P.	M.E.P.	I.H.P.	M.E.P.	I.H.P.	M.E.P.	I.H.P.						
20	153	60	98	44.7	34.08	29.55	18.3	8.46	18.08	14.82	23.6	52.16	41.9	10.26	19.67	365	342
33.3	155	61	75	48.73	61.8	31.28	32.4	6.50	23.1	14.85	39.4	84.9	71.8	13.1	15.44	612	570
50.	152	71	78	53.1	101.1	38.98	60.5	8.03	42.9	15.02	59.9	144.	120.4	23.6	16.37	922	850
66.7	149	85	88	59.	149.8	43.73	90.6	8.54	60.9	13.15	70.10	210.7	160.7	50.	23.71	1230	1118
75.	140	78	94	54.15	154.7	37.52	87.4	7.93	63.5	16.64	99.5	218.2	186.9	31.3	14.34	1383	1249
90.9	145	57	128	49.8	172.3	27.52	77.75	9.76	94.8	19.14	138.6	267.1	216.35	50.75	19	1673	1490
100.	140	73	109	53.2	202.6	26.15	81.2	11.07	118.4	19.16	152.8	321.	234	87.	22.1	1843	1625

NOTE.—The above data are from a total of thirty-six readings taken at various speeds.

To determine the conditions in average operation, the curve at 66 revolutions should be followed (Fig. 96), at which the indicated horse-power of the steam cylinders was 210, and that of the air cylinders, 128. This shows the efficiency to be 61 per cent., the friction loss being 81.5 horse-power, or 39% of that delivered by steam end. This extremely high friction loss was due to the fact that the compressor shaft was out of line, and the plant could not

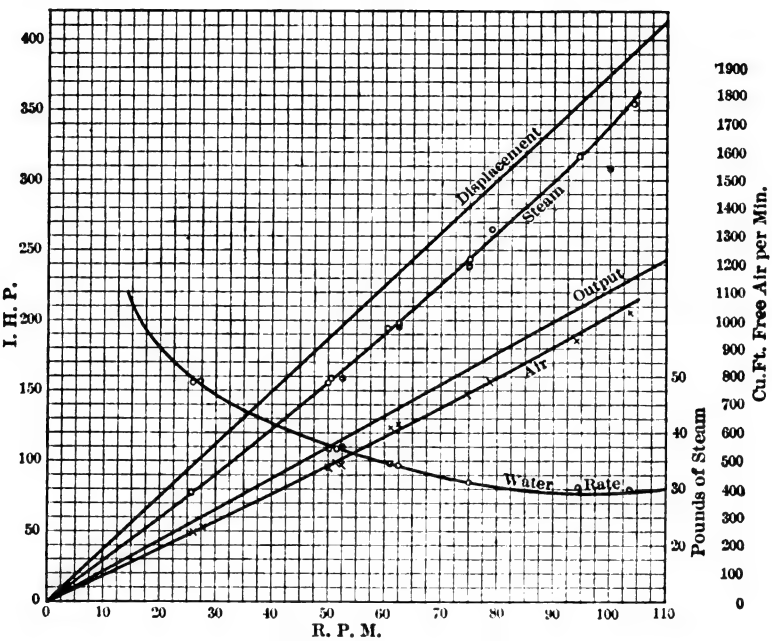


FIG. 96.—Compressor Plant No. 3.

be shut down long enough to rectify it. The details and results of this test, given in Table XI and Figs. 96, 97 and 98, are interesting in exhibiting the inefficiency that may be caused by a purely mechanical defect.

Test Number Four. The results of a test on another plant are given in Table XII and Fig. 99, the details of the boiler test and of the costs being omitted. In this case the compressor was of the tandem compound non-condensing type, with Corliss

valve gear for the steam cylinders. The test shows that, at a low speed, the steam consumption increases more rapidly than with the Meyer type of valve.

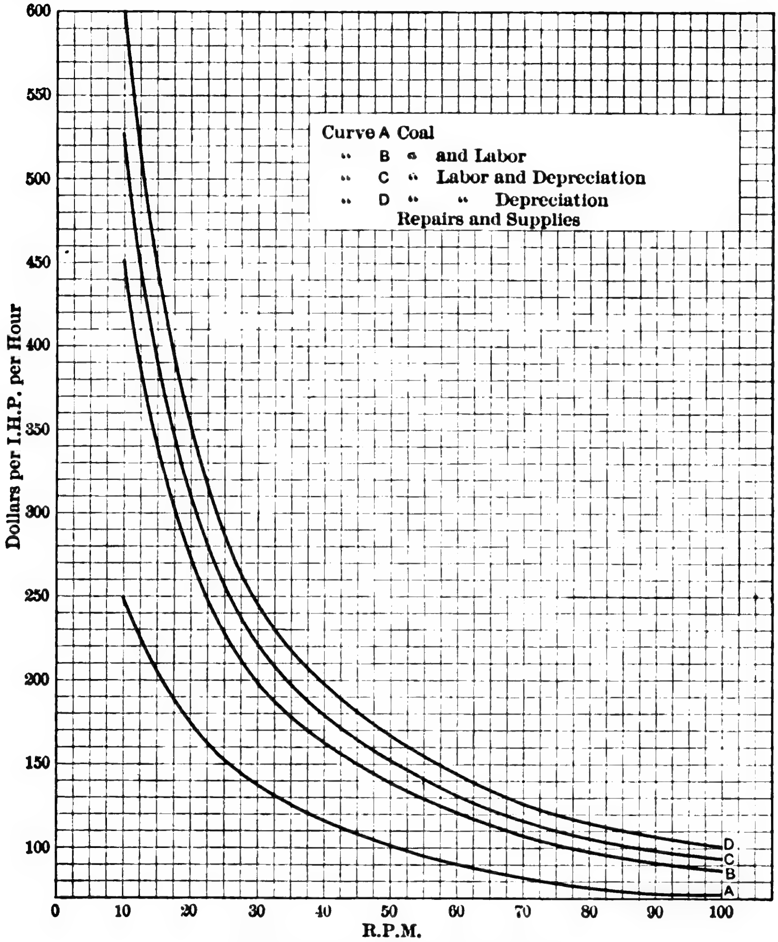


FIG. 97.—Compressor Plant No. 3.

Summary. The results of these tests are enlightening, in showing the actual amount of the losses occurring in the compression of air, particularly when the compressor is operating under the unfavorable conditions of varying air consumption,

necessarily obtaining in mining and other work in which machine drills play an important part. These losses are always recognized as existing, by compressor builders and by intelligent users, and

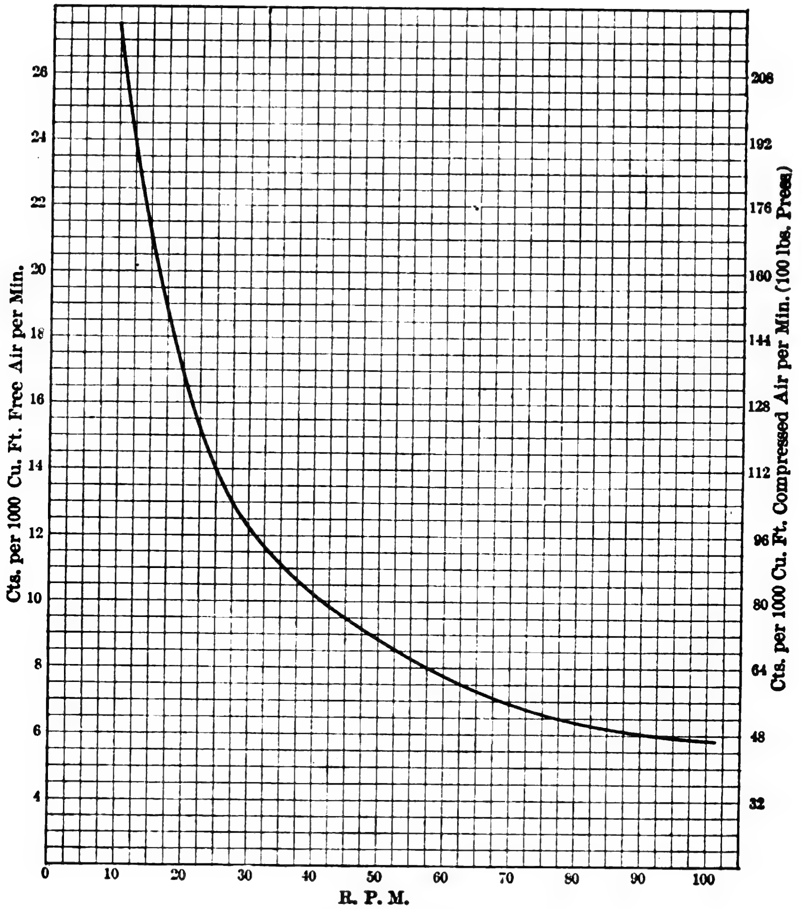


FIG. 98.—Compressor Plant No. 3.

it is clearly desirable that properly conducted tests should be made more frequently.

Again, compressor plants generally develop less power than their full rated capacity. It should be remembered that an air

compressor is essentially a variable speed machine, its speed being regulated by some form of throttling governor, connected with the air-pressure regulator. The machine is therefore called on to run only as fast as the demand for air may require. It may be suggested that it would be well for compressor builders to give in

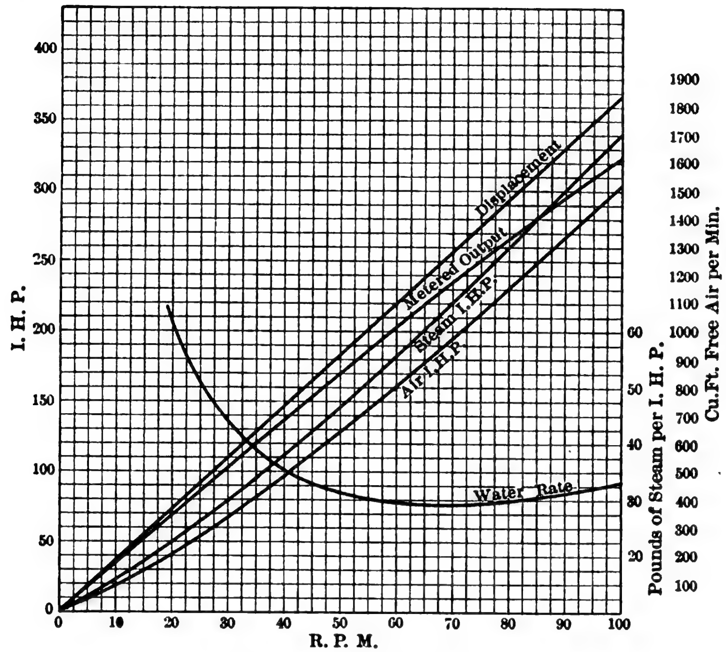


FIG. 99.—Compressor Plant No. 4.

their catalogues the actual horse-power rating at different speeds, with a table of efficiencies at different loads and speeds, just as is done by some of the manufacturers of electrical machinery. Catalogues might also include some definite data respecting the cost per horse-power delivered by the air end of the compressor at different working speeds.

CHAPTER XI

AIR RECEIVERS

On being discharged from the compressor cylinder the air is led into a receiver before passing to the air main. Users of compressed air have been slow to realize the important part played by the air receiver in the economical operation of compressors, and until recently insufficient attention has usually been given to questions of its capacity, design, and position relative to the compressor.

In its common form the receiver consists merely of a cylindrical shell of steel plate, resembling a steam boiler without tubes or flues. It is provided with pipe connections to the compressor and air main, a pressure gauge, safety-valve, drain cock, and man-hole. The receiver may be set vertically or horizontally, the vertical form being generally preferable, as it occupies less floor space (Fig. 100). Another design, which may also be employed as an intercooler, is illustrated in Fig. 56. The cubic capacity of the receiver should be properly proportioned to the size of the compressor. The dimensions range from, say, 24 ins., diameter by 4 or 6 ft. long, up to 48 or 60 ins. by 14, 16, or 18 ft., the largest sizes having a capacity of from 200 to nearly 400 cu. ft. Receivers are usually built to stand a test of 165 lbs. cold-water pressure, for working under pressure of 100 to 120 lbs., higher pressures than this being rarely necessary in ordinary practice, such as mine service. The shells are single-riveted on circular seams and, except for small sizes, double-riveted on longitudinal seams; the heads being dished or hemispherical. To produce the best results, the receiver should be placed close to the compressor, or in any case not more than 40 to 50 ft. distant. A large horizontal receiver is shown in Fig. 101.

The principal functions of an air receiver may be summarized

as follows: (1) to eliminate the pulsating effect of the strokes of the compressor piston and prevent rapid fluctuations of pressure; (2) to minimize the frictional loss attending the flow of air through the lines of piping; (3) to serve in some degree as an equalizer and reservoir of power; (4) to cool the air as thoroughly as possible before it passes into the main, thus causing it to deposit a part of its moisture in the receiver, whence it is drained off.

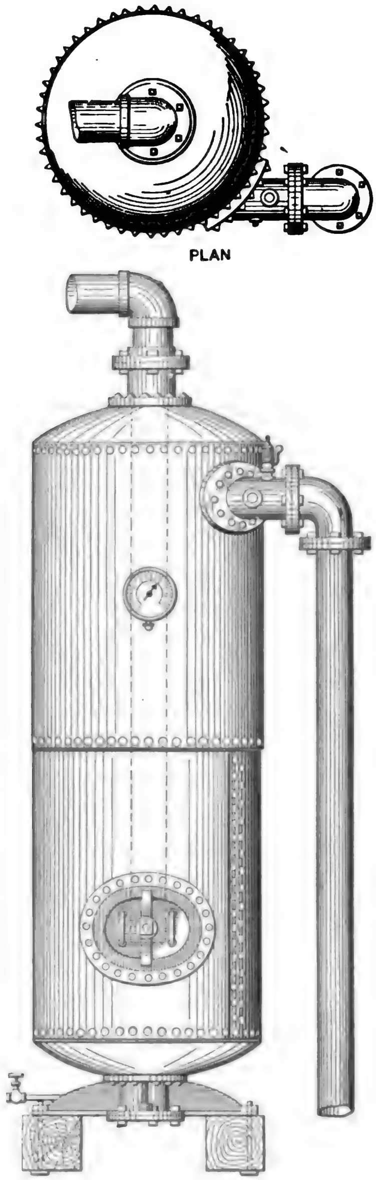


Fig. 100.—Vertical Air Receiver (Norwalk Iron Works Co.).

Regarding the first point, the volume of the receiver should be sufficiently great in proportion to that of the compressor cylinder to prevent any material rise of pressure in the receiver by the incoming volume of air forced into it at each stroke. If the compressed air were discharged directly into the main, large fluctuations of pressure would occur, accompanied by periodic acceleration of flow. This would not only increase the frictional resistance in the pipe, but at the end of each stroke the compressor piston would have to force the air out of the cylinder against a pressure momentarily greater than the normal. A loss of power would thus be caused, and the variation of the

work done throughout the stroke of the piston would be increased. The violence of the discharge pulsations is obviously greater with a single cylinder than a stage compressor, working to the same air pressure, because the total discharge must take place from the cylinder of larger diameter in a smaller proportion of the length of stroke than is the case with the high-pressure cylinder of a stage compressor. In the latter the delivery valves open earlier in the stroke, and the air pipe is about one-half the diameter of the cylinder.

FIG. 101.—Horizontal Receiver-Aftercooler (Ingersoll-Rand Co.).

The second function of the receiver is best fulfilled by placing an auxiliary receiver near the point at which the compressed air is used. Just as the receiver at the compressor diminishes the momentary rise of pressure in the main caused by each stroke of the piston, so a second receiver close to the engine or machine using the air will prevent a drop of pressure as each cylinderful of air is drawn off. By reducing the fluctuations of pressure the two receivers maintain a practically constant flow of air through the main connecting them and the friction and loss of pressure are thus minimized.*

For mine service the second receiver would usually be placed somewhere underground. This arrangement is always advantageous when the air main is of great length. Underground receivers are not often used for air drills alone, but they become a

* Other questions relating to the flow of air in pipes, frictional losses, etc., are discussed in detail in Chapter XVI.

necessity when large machines, such as pumps and hoists, are run by compressed air. They are useful, moreover, in permitting a further deposition of moisture from the air, thus rendering the air dryer and more suitable for use in expansive-working engines. To be most effectual in accomplishing this, the underground receiver should be placed at the point in the pipe line where the air has reached its lowest temperature—a consideration not always consistent with the local conditions.

Underground receivers are usually similar in construction to those installed near the compressor. Sometimes, however, as for example at the Mansfeld copper mines, Germany, another mode of construction has been satisfactorily adopted. A chamber is excavated in the rock, all loose stone removed, and the walls cemented tight. The chamber is closed by a brick dam composed of two parallel walls, with a two-inch layer of cement between them. In the dam are set a cast-iron man-hole with suitable cover, several pipes for connecting with mains to the various working places, and a drain pipe and cock close to the floor. The latter is opened from time to time, to blow out the accumulated water and sediment. A pressure gauge is attached to the man-hole cover. Such reservoirs may be built to cost much less (for large sizes) than ordinary shell receivers of equal capacity.*

The third function of the receiver is apt to be misunderstood or exaggerated. While it is true that it acts to a limited extent as a reservoir of power; yet, to be of much practical service in this respect, its capacity must be very large.

For example, take a 20-in. compressor, working at 60 lbs. pressure to supply air for a regular consumption. To enable the receiver to meet the demand for only 1 minute after the compressor is stopped, and not have the pressure fall more than 15 lbs., it would have to be 5 ft. diameter by 50 ft. long. Again, if the compressor were running at a constant speed and the demand for air should suddenly increase 25 per cent.—as might happen in

* *Zeitschrift für das Berg-, Hütten- und Salinen-Wesen*, Vol. XLI, p. 119. A receiver of the kind mentioned was built at Mansfeld for about one-third the cost of an equivalent steel receiver.

starting several more machine drills—a receiver of the size mentioned could meet the extra demand only 4 minutes.* It is thus evident that while a receiver is useful as an equalizer within certain limits, yet, unless it be large, the pressure might quickly run up to an unreasonable amount in case of an unexpected decrease in consumption of air. Long pipes of large diameter assist in equalizing the flow of air, but their use does not preclude the necessity of receivers. It is much cheaper to employ piping of moderate size, in connection with a receiver of generous dimensions.

Probably the most important office of the receiver is to cool the air before it passes into the main. In recent years much more attention than formerly has been given to this point. The velocity of flow of the air coming from the compressor is greatly reduced on entering the relatively large volume of the receiver; it is cooled somewhat at the same time, and caused to deposit a part of the moisture in suspension, which otherwise would be conveyed into the system of piping, and thence to the machines using the air. It is intended that the receiver shall be of sufficient capacity to drain the air as thoroughly as is economically practicable. But in the ordinary sizes of shell receiver the results are usually quite imperfect, because the air passes through too rapidly to permit any large drop in temperature. The inlet and outlet pipes of the receiver should be placed in proper relative positions. If at opposite ends, and especially if these pipes point toward each other, a strong through current is caused, which reduces the usefulness of the receiver. A large part of the entering volume of air passes out again without having had time to cool or to drop much of its entrained moisture. One mode of arranging the pipe connections is to place the inlet on one side, near the end of the receiver, while the outlet is at the opposite end, in the middle of the head. The air is thus forced to change its direction of flow. Or, as in Fig. 100, both pipes may be connected near the top, the outlet pipe being carried through the receiver nearly to the bottom, where the air is likely to be slightly cooler (and dryer) than at the top. As the inlet pipe shown in this

* Norwalk Iron Works Catalogue, 1906, p. 63.

case is connected tangentially to the periphery of the receiver, a rotary motion is imparted to the body of air, so that each particle remains longer in the receiver and under its cooling influence. Some receivers are provided with baffle-plates for the same purpose, as in Fig. 101. With wet compressors a large amount of moisture is carried into the receiver; even in dry compressors some water collects from the natural moisture of the atmosphere, especially in warm weather. Part of the lubricating oil carried over from the compressor cylinder is also deposited in the receiver. At intervals, according to atmospheric and other conditions, the water and oil are drained off by means of the cock provided for the purpose.

Another result of cooling in the receiver may be noted. A receiver of ample size, placed close to the compressor, tends in some degree to economize power; because, whatever cooling is accomplished reduces proportionately the temporary increase of pressure due to the heat of compression. Hence, the piston consumes somewhat less power in forcing the air out of the cylinder against the receiver pressure than if the air were left to cool gradually in a long length of piping. As the heat of compression must be lost in any case before the air is used, this saving is worth while, however small it may be, since it is produced without cost and incidentally to the normal operation of the receiver.

This consideration has of late led to the employment of what are called "receiver after-coolers," (Fig. 101), practically identical in construction with the large tubular intercoolers shown in Figs. 54 and 56.* The shell contains a series of water-cooled tubes, between which the air is caused to circulate before passing to the larger outer portion of the receiver, whence it is discharged into the main. Having a sufficient volumetric capacity and cooling area of tubes, this type of receiver cannot fail to be more efficient as an after-cooler and the benefits of employing a receiver are more fully realized.

* These are referred to, in the latter part of Chapter VI, as being applicable as intercoolers for stage compressors. See also an article by Frank Richards, in *Compressed Air*, Jan., 1907, p. 4329.

CHAPTER XII

SPEED AND PRESSURE REGULATORS FOR COMPRESSORS

If the consumption of compressed air were constant, no more regulation of the compressor's speed and power would be required than that furnished by an ordinary governor for the steam end, to take care of fluctuations in boiler pressure or accident to some part of the mechanism. But the conditions under which most air compressors operate make it necessary to provide for running economically even when there are wide variations in the rate at which the air is used. In event of a sudden temporary decrease in consumption, the compressor must be slowed down, the alternative being to blow off air at the receiver safety valve, just as steam would be blown off in similar circumstances from a boiler. As a cubic foot of compressed air, however, costs more than a cubic foot of steam, the air cannot be allowed to go to waste at a safety valve. The compressor must be furnished with some device for coördinating the quantity of steam admitted to the steam end with the variable air pressure in the receiver, thereby regulating the piston speed in accordance with the demands upon the air end. Furthermore, it is not enough to provide only for varying the speed of the compressor. At times, the consumption of air may cease entirely for a short period, and, to avoid the necessity of bringing the compressor to a standstill, provision should be made for unloading the air end. When this is done useful work stops for the time being, the compressor consuming only enough steam to overcome friction of the moving parts, and turn its centers.

Numerous regulating and unloading mechanisms have been devised, so that instead of requiring the almost constant attendance of an engineer at the throttle, the modern air compressor operates

automatically under the widest variations of load. As these useful devices differ greatly in design, the subject will best be illustrated by giving a few examples in detail. They may be classified under two heads: (1) speed governors and pressure regulators; (2) unloaders for the air cylinders.

Speed Governors and Pressure Regulators. Speed governors are usually of the ordinary centrifugal or fly-ball type, and may be

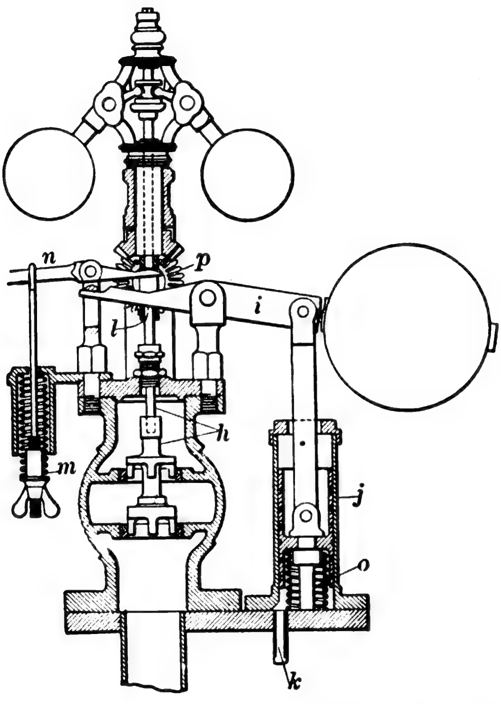


FIG. 102.—Clayton Governor and Pressure Regulator.

applied to the steam end of the compressor merely to regulate its speed, as in case of a steam engine; or their action may be so modified and controlled by the changing receiver pressure as to produce a combined speed and pressure regulation. The air cylinder is not completely unloaded at any time, the compressor being simply

speeded up or slowed down in conformity with the rate at which the air is used.

The pressure regulator of the fly-ball governor type may be illustrated by Fig. 102 (Clayton governor). The stem of the throttle valve, *h*, which is inserted in the steam pipe, connects with the spindle of the ball governor, by which the speed of the compressor is limited and controlled. At *p* is shown the bevel gearing for operating the governor, a small pulley being mounted on the gear shaft and driven by belt from the crank-shaft of the compressor. By means of the weighted lever, *i*, and the small air cylinder, *j*, the action of the ball governor is modified by the air pressure in the receiver. Air from the receiver enters the cylinder, *j*, through the pipe, *k*, and when the pressure exceeds its assigned limit, raises the piston and weight, and shuts off steam by forcing down the throttle valve, *h*, the pressure of the lever being applied at the point, *l*. The governor may be adjusted to its work by the spring and thumb-screw, *m*, acting on the small lever, *n*, which tends to keep open the throttle against the downward pressure of the weighted lever, *i*, upon the valve stem. The spring, *o*, is introduced to ease the drop of the weight when the air pressure falls.

Other designs, similar in general principle but varying in many details, are used on the Ingersoll-Rand, Sullivan, Franklin, McKiernan, American, and other compressors, when steam-driven and of the straight-line or duplex type. The Sullivan speed and pressure regulator, as supplemented by an unloading attachment, is described hereafter.

An entirely different form of governor is the Norwalk (Fig. 103). A balanced throttle valve, *a*, is placed in the main steam pipe, and above it is set a small air cylinder, *b*, the piston rod of which is a prolongation of the valve stem, *c*. At the side of the cylinder, *b*, is a spring safety valve, *d*, connected by a pipe, *e*, with the receiver, or with the air main leading to it. By means of a hand-wheel, *f*, on the safety valve, the spring is adjusted so that the air will lift the valve, and pass through it, at any desired pressure. When the receiver pressure exceeds this limit the safety valve, *d*, rises and allows air to pass under the piston in the small cylinder,

b, raising it and partly closing the throttle. If no escape were provided the piston would be forced at once to the top of the cylinder. To regulate its movement and prevent shutting off the steam completely, a very narrow vertical slot is cut in the side of the cylinder. As the piston rises, more and more of this slot is uncovered and furnishes an escape for the air passing into the cylin-

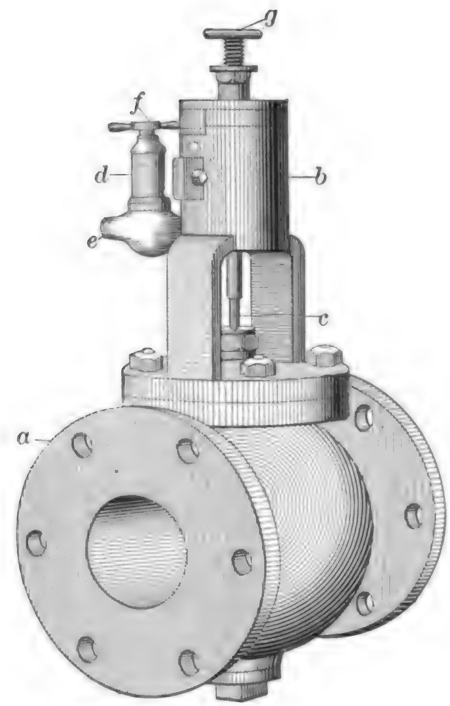


Fig. 103.—Norwalk Pressure Regulator.

der. The slot being very narrow, a slight difference in the quantity of air causes the piston to assume a high or low position. In this way the throttle is moved, controlling the admission of steam and the compressor speed. As the air pressure falls the valve begins to open again. To prevent the small piston from rising too far and stopping the compressor by completely closing the throttle, a screw stop, *g*, is set in the top of the regulating cylinder,

b. This can be so adjusted by hand that, when the small piston has reached the top of its stroke, just enough steam is admitted by the throttle to keep the compressor in motion.

In another form of this governor, shown in vertical section, Fig. 104, the fine slot in the little cylinder, B, is replaced by a tapered

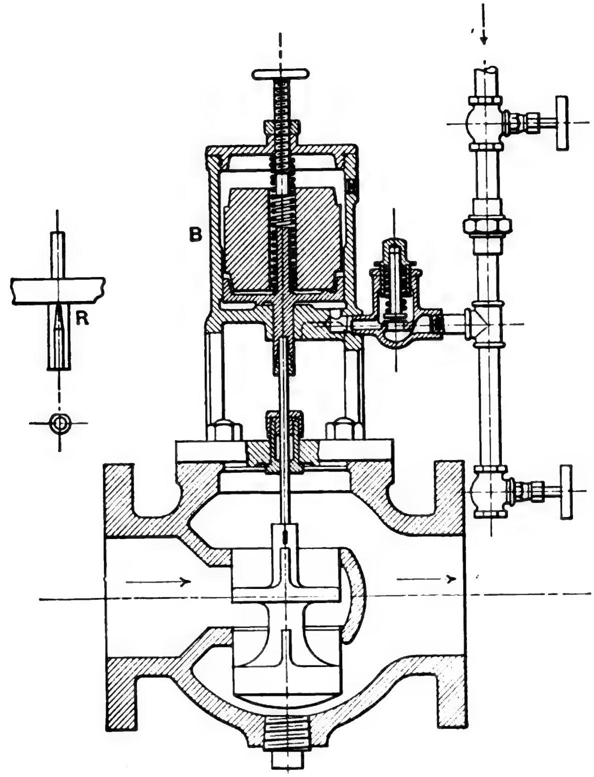


FIG. 104.—Norwalk Pressure Regulator.

recess in the stem or piston rod of this cylinder, at the point where it passes through the lower head (indicated at R, in the small cut to left of main figure). As the piston in this cylinder is forced upward by the air pressure the area of the opening formed by the slotted stem furnishes a graduated escape for the air, and so

regulates the small piston's movement, and through it the throttle valve.

The Ingersoll-Rand Co. also makes a steam regulator in which the stem of the main throttle is prolonged to the piston of a small horizontal air cylinder, attached to the side of the throttle. This piston is moved by air pressure conveyed through a quarter-inch pipe from the receiver.

Another governor (Clayton) is shown in Fig 105. The throttle valve, *a*, is provided with a lever and weight, *b*, connected

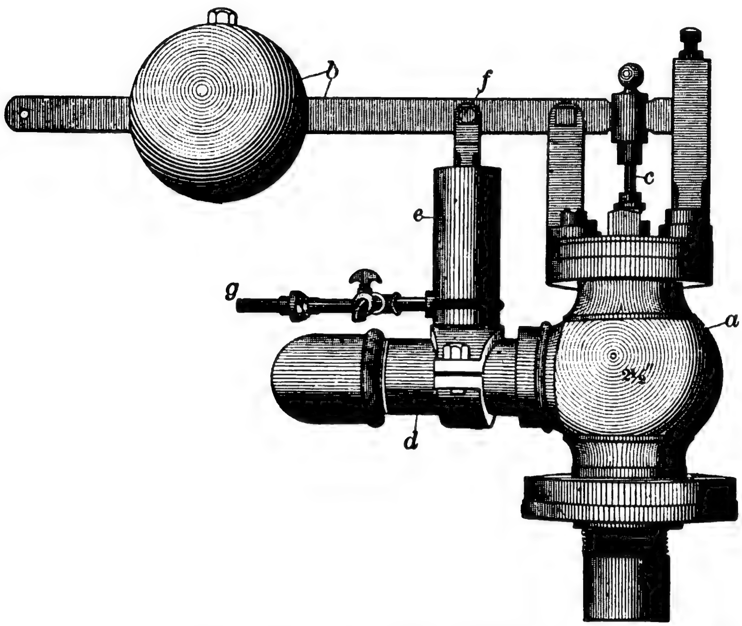


FIG. 105.—Clayton Pressure Regulator.

with the valve stem, *c*. Close alongside of the throttle, and for convenience clamped on the steam pipe, *d*, is a small air cylinder, *e*, the upper end of whose piston rod is pinned at *f* to the weighted lever. Entering the bottom of this air cylinder is a small pipe, *g*, from the air receiver. When the pressure in the receiver exceeds the assigned limit the weighted lever is raised and, partially or wholly, closes the steam throttle, *a*. Then, when the air press-

ure has been reduced by the slowing down of the compressor, and by consumption of air from the receiver, the weight falls and reopens the throttle.

With governors and regulators of the type represented by the preceding examples, the operation and control of the compressor is not automatic under all conditions, but they answer the purpose for some kinds of service. In case no air is drawn from the receiver, the compressor slows down until it just passes its centers; then, in most of the designs, if the pressure continues to rise, a little air will blow off at the receiver safety-valve, or the compressor may be stopped completely by closing the steam throttle.

Air-Cylinder Unloaders. These are designed to exercise complete automatic control over the compressor, when the latter is belt-driven; and also for steam-driven compressors when used in conjunction with a governor. The throttle is first nearly closed (in steam compressors), as the consumption of air decreases; then, if it ceases altogether, the unloading mechanism either shuts off the intake air or else opens the discharge valves, thus admitting air at receiver pressure to both ends of the cylinder. In either case the pressures on opposite sides of the piston are balanced and all useful work ceases, though the compressor continues to turn its centers, taking only enough steam to overcome friction.

The Rand "Imperial" unloader, for small compressors driven by belt or direct-connected electric motor, furnishes an example of this type of regulator (Fig. 106). It is inserted in the intake pipe, and shuts off the air from the inlet valves when the receiver pressure rises above the set limit. In the cut the passage of the intake air is shown by the arrows. The small chamber (60) is connected by a ¼-in. pipe with the receiver. As the pressure increases, the piston (57) moves against the resistance of the spring (56), admitting receiver air, through the small ports on the left of the piston, to the lower side of the plunger valve (61). On reaching its seat this plunger closes the intake to the compressor cylinder. The resistance of the spring (56) may be

adjusted by the screw-plug (55), for any required working pressure. As the receiver pressure falls again, on increased consumption of air, the spring forces down the piston (57). This closes the lower small air port, leading to the under side of the

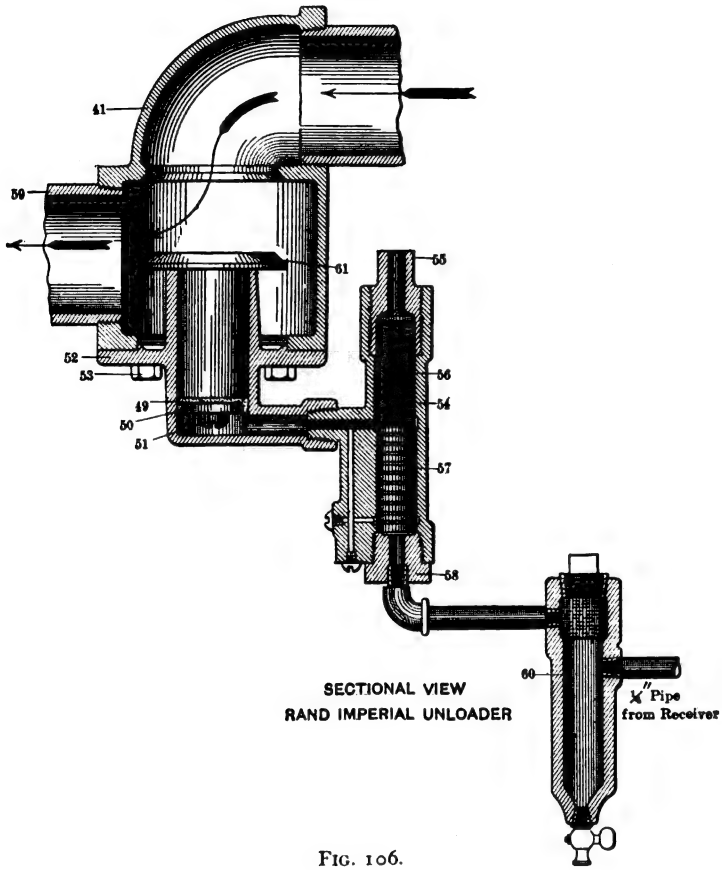


FIG. 106.

plunger valve (61), and at the same time opens the upper horizontal port, connecting with the open screw-plug (55). The air below the plunger valve is thus exhausted, causing the latter to fall, thus reopening the intake passage. The useful work of the compressor is then resumed.

An unloader similar to the above is used for some of the Allis-Chalmers compressors. The Ingersoll-Rand Co. makes an automatic "choking" controller, which is applied to the intake pipe of the piston-inlet compressor. It is adjustable for any desired limit of pressure by a weighted lever, and may be used for all forms of steam-driven compressors.

A recent form of this unloader is shown on the folding plate, Fig. 33, as applied to one of the latest designs of the Ingersoll-Rand compressors—"Imperial," type 10. The same company is now making a clearance or expansion controller, for unloading the air end of the compressor. It varies the clearance volume of the cylinder by automatically varying the number of discharge valves in action. A small air cylinder is placed between the main cylinders and connected with the receiver. As the receiver pressure increases, the piston of the controller cylinder rises higher and higher. Inserted in the side of this cylinder is a series of small pipes, each connected by branches with a discharge valve on opposite ends of the main cylinder. These valves are thus released from the receiver pressure successively, as the pressure increases, and the work done by the compressor is proportionately reduced.

A combined governor and pressure regulator, with unloading attachment, as employed by the Sullivan Machinery Co., will illustrate a type of compressor regulator that has been adopted by several builders, though with many variations in details of design (Fig. 107). It may be used with straight-line or duplex, steam-driven compressors. The split-ball governor (11), belt-driven from the crank-shaft to the pulley (20), accompanied by the tightener (43), controls the steam throttle (3). Connected with the governor spindle and throttle valve stem, at 28, is a lever (25), the position of which is influenced by the centripetal action of the set of springs (31, 32, and 26). By screwing up or down the hand-wheel and speeder screw (5), this system of springs (and with them the governor) is set to run the compressor at any desired speed. The other element of the governor is the air-pressure device, which, by the position of the plunger in the

small air cylinder (18), causes the springs to be brought into action in the order of their strength, thus producing movement of the lever (25).

The pressure device is connected with the air receiver by the union valve (33), admitting air to the little cylinder (27), the piston

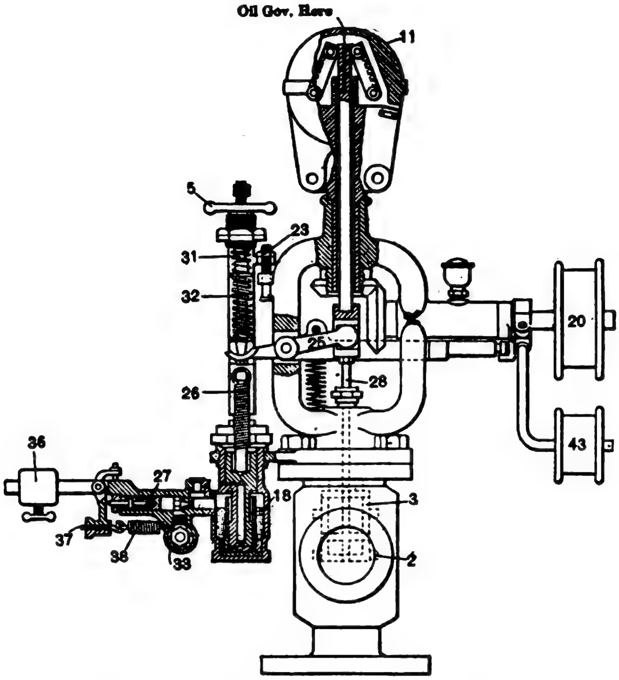


FIG. 107.—Sullivan Governor and Unloader.

of which operates a needle valve. This valve is held closed against any desired minimum air pressure by the adjustable weight (36) and the regulating screw and spring (37 and 38). When the receiver pressure rises above the normal, it opens the needle valve and admits receiver air to the cylinder (18). As the pressure increases, the plunger in (18) rises against the counter-spring (26) and through the lever (25) tends to close the main steam throttle (3), thus slowing the compressor. Total stoppage

is prevented by screwing down the nut of the stop-screw (23), so as to limit the upward movement of the pressure plunger. This plunger is designed to act intensively, being so proportioned that a variation of only 2 or 3 lbs. receiver pressure is multiplied to 40 lbs. in its action on the governor. In this way a sensitive control is produced within narrow limits of working air pressure. To prevent violent movements of the pressure ele-

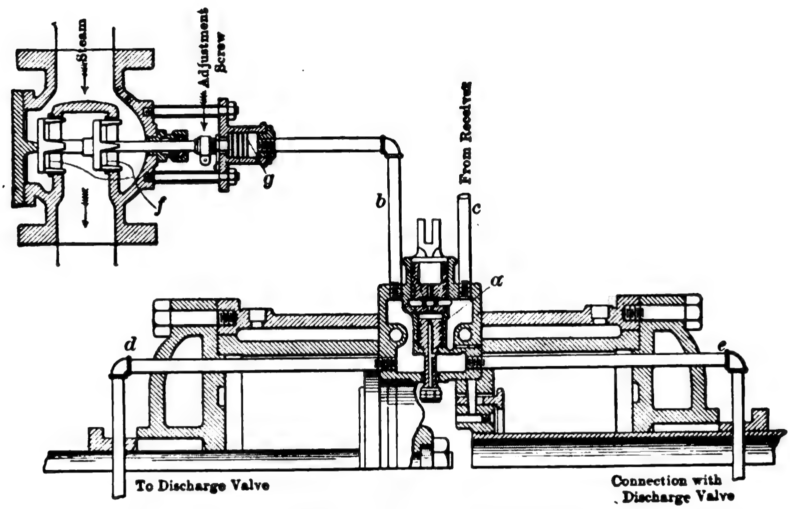


FIG. 108.—Ingersoll-Sergeant Regulator and Unloader.

ment, in case of sudden changes of receiver pressure, the plunger in (18) is provided with an oil dash-pot.

A somewhat similar pressure regulator and unloader is used on the Franklin compressor.*

Another unloader, applicable to straight-line and duplex compressors, and in a modified form to stage compressors also, is the Ingersoll-Sergeant. It differs materially from the unloaders previously described, in controlling the action of the discharge, instead of the inlet valves. The principles of its construction and operation will be understood by reference to the longitu-

* *Mines and Minerals*, May, 1905, p. 504.

dinal section in Fig. 108. The most recent design of this unloader differs in some details from that shown in the cut, but its mode of working is unchanged, and many are in use.

A weighted plunger, *a*, working in a small cylinder, is attached for convenience to the shell of the compressor cylinder. From the chest in which *a* is set there are four pipe connections as shown: *b* leads to a balanced throttle valve in the main steam pipe, *c* connects with the air receiver, and *d* and *e* with one or more discharge valves at each end of the cylinder. The stem of the steam throttle, *f*, is a continuation of the piston rod of a small horizontal air cylinder, *g*, which is attached to the side of *f*. Behind the piston of this little cylinder enters the air pipe, *b*. When the pressure in the receiver becomes too great the safety valve, *a*, rises, and exhausts the air behind the two discharge valves which are connected with the pipes, *d* and *e*. This admits air at receiver pressure into each end of the compressor cylinder, thus balancing the pressure on the two sides of the piston and unloading the engine. At the same time the air in the little cylinder, *g*, is also exhausted, so that the throttle valve, *f*, moves to the right and admits only enough steam to keep the compressor slowly turning. When the compressor is thus unloaded no work is done; the air is merely circulated through the pipes, *d* and *e*, from one end of the cylinder to the other, until more air is drawn from the receiver and the pressure reduced. Then the safety valve, *a*, closes and the pipes, *d* and *e*, are again filled with compressed air. The steam throttle is also forced open by the pressure through the pipe, *b*, and compression goes on regularly. The admission and discharge lines of an air card from a compressor thus unloaded form practically a single horizontal line, at a height above the atmospheric line representing the receiver pressure.

For steam-driven compressors of the Corliss type, as built by the Ingersoll-Rand, Nordberg, Laidlaw-Dunn-Gordon, Sullivan, Allis-Chalmers, and some other companies, the air-pressure regulators act in conjunction with ball or other centrifugal governors. All of them control the operation of the compressor

by acting upon the expansion gear of the steam end and changing the point of cut-off.

The Laidlaw-Dunn-Gordon governor (Fig. 109) may be taken as an example. Air is admitted from the receiver to the small cylinder, A, the piston of which is weighted, as shown. The

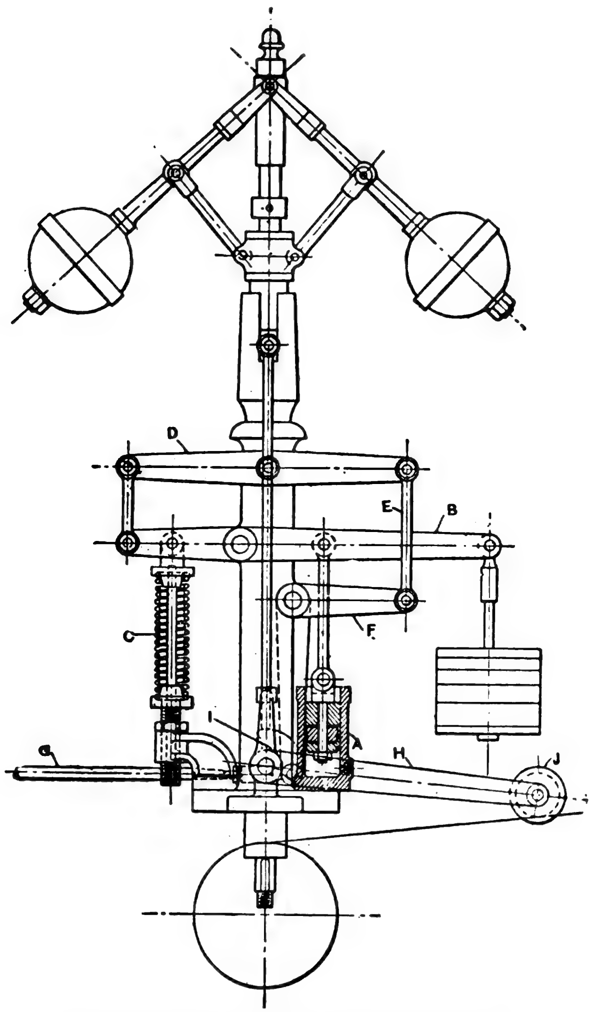


Fig. 109.—Laidlaw-Dunn-Gordon Air Governor.

action of the lever, B, supporting the weight is adjusted by the coil spring, C. This lever is linked to a floating lever, D, pinned to the vertical side rods of the ball governor. D is connected by the link E to the bell-crank, F, the lower arm of which is pin-connected to the long horizontal rod, G. By this system of levers, the movement of G, and through it the point of cut-off of the Corliss steam gear, is under the combined control of both ball governor and of the receiver pressure as influencing the position of the piston of the cylinder A. The arm, H, is pivoted at the foot of the governor post. Connected to it are the cam, I, and the idler pulley, J, which rests on the governor belt. In case the belt breaks, the idler pulley falls and the cam allows the governor to drop, thus shutting off steam and preventing the compressor from racing. The designs of governors of this type are worked out in a number of different ways.

Another example of governor is that employed on the constant-speed, variable-delivery compressor, built by the Nordberg Manufacturing Co. It is for motor-driven machines, with Corliss air valves, and operates by closing the inlet valve before the stroke is completed.* During the remainder of the forward stroke, the air already admitted to the cylinder is expanded below atmospheric pressure, and is then compressed on the return stroke. This is practically equivalent to varying the working length of the stroke.

A general view of this compressor is shown in Fig. 110, and the construction of the valve gear is indicated in Fig. 111. The wrist-plate *w* is driven by the rod *a*, from an eccentric on the fly-wheel shaft; another eccentric drives the releasing mechanism, through the rod *b*, which oscillates the arm *c* about the fixed center *d*. Swivelled to the lower end of *c* is a 3-armed rocker. The arm *i* is linked by the rod *j* to the radius fork *k*, which in turn is connected to the pressure governor *l*. The arms *g* and *h* of the rocker, through the rods *e* and *f*, operate the releasing cams *n* and *o*, which are attached to the forward and back inlet-valve spindles. When the compressor is working regularly, under

* *American Machinist*, Aug. 22d, 1907.

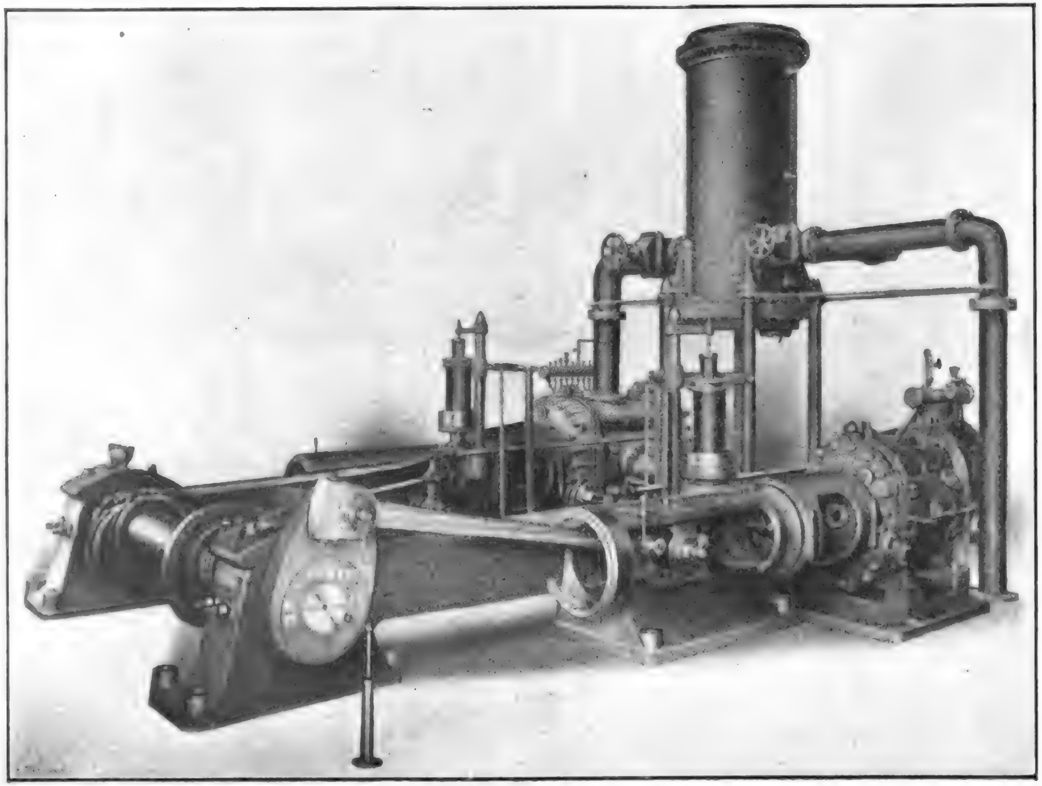


FIG. 110.—Nordberg Constant-Speed, Variable-Delivery Air Compressor.

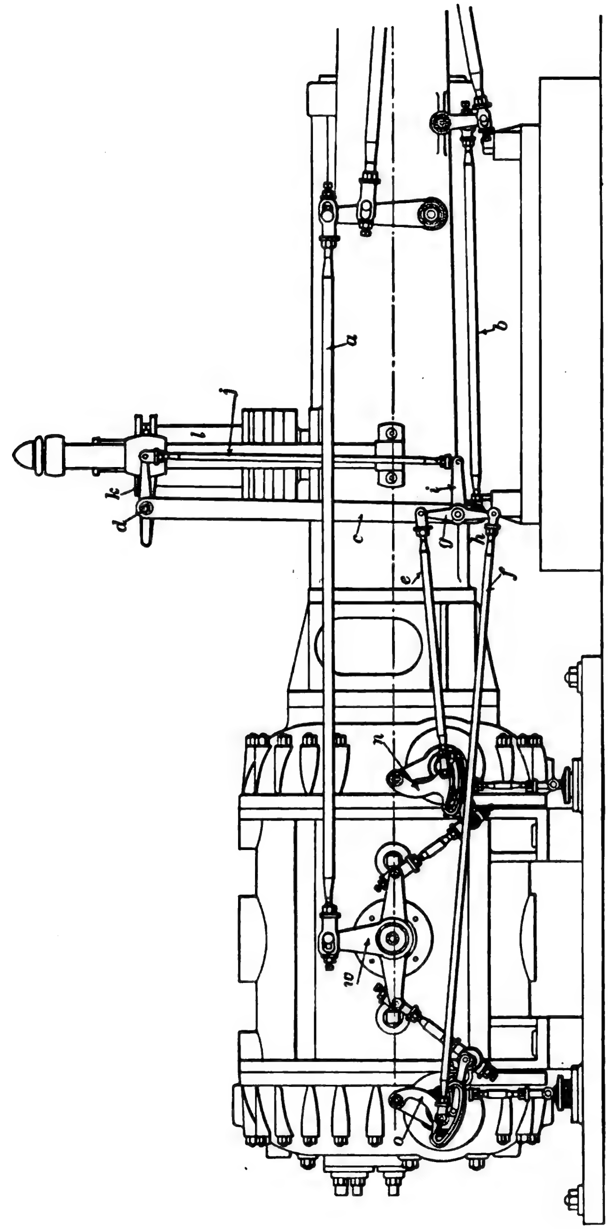


FIG. 111.—Valve Gear of Nordberg Constant-Speed, Variable-Delivery Air Compressor.

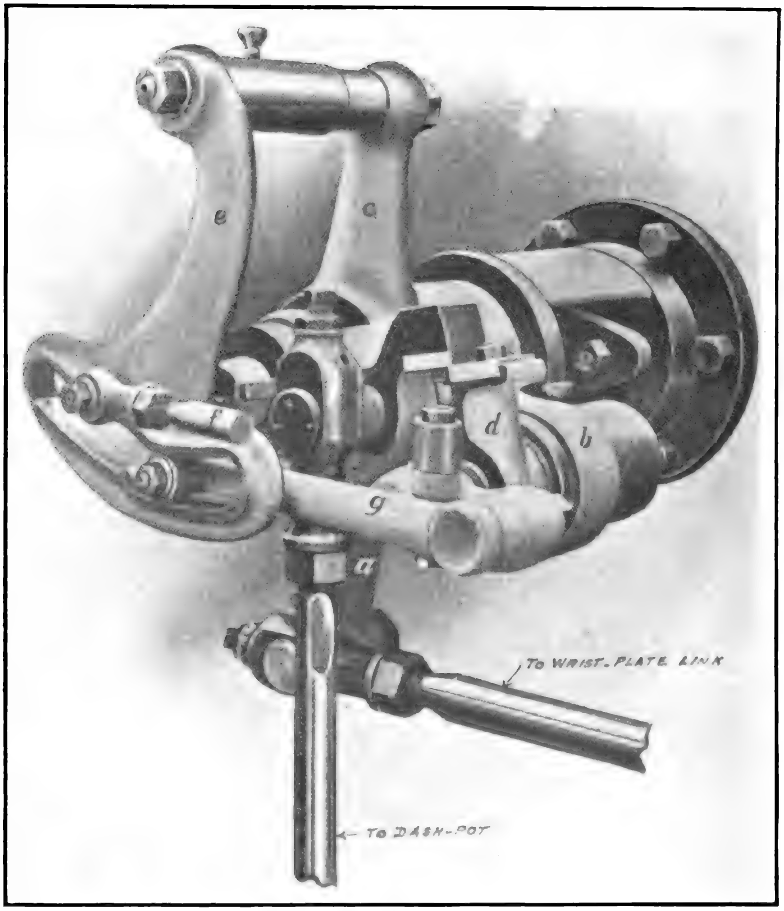


FIG. 112.—Detail of Valve Gear Shown in Fig. 111.

normal consumption of air, the rocker arms *g* and *h* maintain a vertical position, under the action of the eccentric rod *b* and impart equal movement to both knock-off cams. If, however, the receiver pressure increases, the rocker arm *i* will be drawn upward, the arms *g* and *h* will take an inclined position and, through the rods *e* and *f*, the point of release of the valves is altered.

The releasing mechanism is shown by Fig. 112. Mounted on the valve spindle is a rocker having three arms, *a*, *b*, and *c*. The wrist-plate link is connected to *a*, the releasing latch *d* to *b*, and the governor cam-arm *e* to *c*. Part *e* is connected also to the governor by the rod *f*, as explained above, and hence has a compound motion: it swings bodily about its swivel pin at the top, and its position is adjusted laterally by the action of the governor. The cam slot has two circular arcs, struck from the center at the upper end of *e*, with an inclined jog connecting them. Since the roller on the arm *g* swings about its center under the action of the cam groove, as the cam is moved from the main eccentric by the rod *f*, the latch *d* is alternately released and engaged, when the roller passes the jog in the cam. The point of the stroke at which the release takes place is determined by the governor, as already stated.

In Figs. 113, 114 and 115, are given a set of indicator cards from a two-stage compressor provided with this regulating mechanism, and running at a speed of 74 revolutions per minute. The upper card in each cut is from the intake or low-pressure cylinder, the lower card from the high-pressure cylinder. Fig. 113 shows the cards when working at nearly full load. Fig. 114 (half load) illustrates the action of the regulating gear. Taking the crank-end card, *C*, the inlet valve remains open from the beginning of the stroke, at *a*, approximately to mid-stroke, *b*, at which point the releasing gear acts and the valve closes. From *b* to the end of the stroke, at *c*, the air in the cylinder expands below atmospheric pressure. On the return stroke, the compression line nearly coincides with the expansion line from *c*, until atmospheric pressure is reached at the point *b*, after which compression proceeds in the usual manner. The action of the inlet

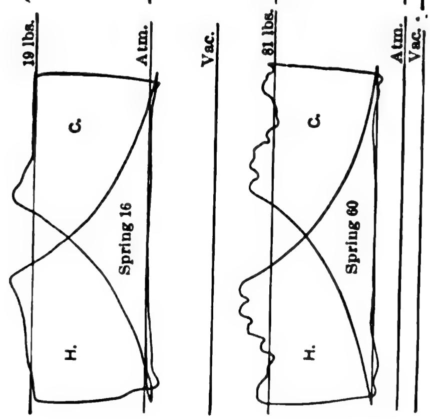


FIG. 113.—Nearly Full Load.

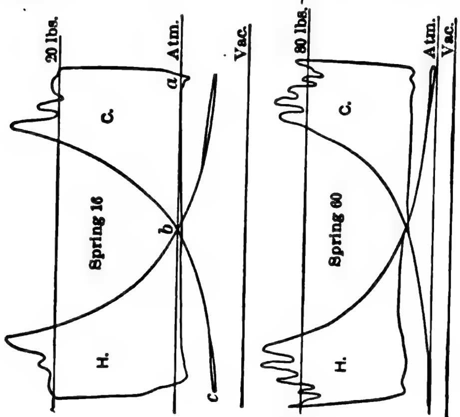


FIG. 114.—Half Load.

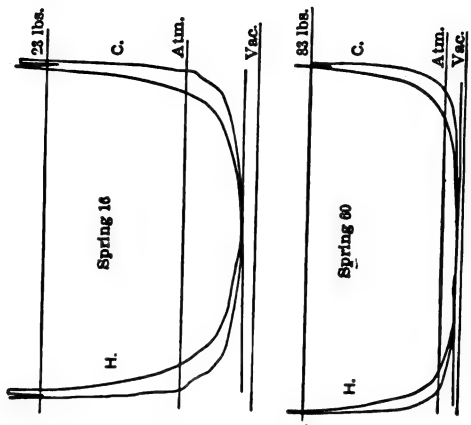


FIG. 115.—Nearly Zero Load.

valves of the high-pressure cylinder is the same, except that the expansion and re-compression of the air is from receiver pressure, instead of atmospheric pressure. In Fig. 115 the cards show the very small amount of work done when the compressor is operating under nearly zero load. To simplify the mechanism, each cylinder is provided with its own pressure governor.

CHAPTER XIII

AIR COMPRESSION AT ALTITUDES ABOVE SEA-LEVEL

BECAUSE of the diminished density of the atmosphere, air compressors do not produce the same results at high altitudes as at sea-level. Their effective capacity is reduced because a smaller weight of air is taken into the cylinder at each stroke. It is necessary, therefore, to modify the figures relating to the capacity and performance of compressors, as set forth in the first part of Chapter X. This matter is of special importance in connection with mining operations, because of the large number of mines situated in elevated mountain regions. The rated capacities of compressors, in cubic feet of air, as given in the makers' catalogues, are for work at normal atmospheric pressure, and due allowance must be made for decreased output at elevations above sea-level. This reduction in output, which is usually also tabulated in handbooks and catalogues, should receive due consideration in order to avoid serious errors. For example, the volume of compressed air delivered at 60 lbs. pressure, at 10,000 ft. elevation, is only 72.7 per cent. of the volume delivered at the same pressure by the same compressor, at sea-level. In other words, a compressor which at sea-level will supply power for 10 rock-drills, will at an elevation of 10,000 ft. furnish air for only 7 drills.

The foregoing statement relates only to the volumetric capacity of the compressor. It must be remembered that the heat of compression increases with the ratio of the final absolute pressure to the initial absolute pressure. As this ratio increases with the altitude, more heat will be generated by compression to a given pressure at high altitudes than at sea-level. This additional heat

temporarily increases the pressure of the air in the cylinder, while under compression, and more power is therefore required to compress and deliver a given quantity of air. The corresponding loss of work, due to the subsequent cooling of the air in receiver and piping, also increases with the altitude.

Contrary to a common impression, the volume of air delivered by a given compressor does not bear a constant ratio to the

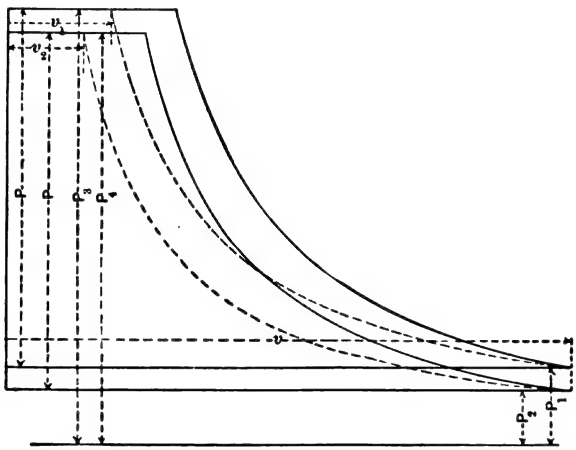


FIG. 116.

barometric pressure, but at different altitudes this volume decreases slower than the barometric pressure. This relation may be shown as follows.* Two ideal indicator cards are represented in Fig. 116 one of a compressor working at sea-level, with an initial pressure P_1, the other at an altitude with a lower initial pressure P_2. The initial volume V and the final gauge pressure P, are the same for both compressors, P_3 and P_4 being the respective final absolute pressures. V_1 and V_2 are the final volumes, corresponding to the dotted isothermal curves, these volumes being taken as the basis, because they are those to which the com-

* The general method of demonstration here given, together with Fig. 116 and accompanying table, are taken by permission from an article by F. A. Halsey, in *American Machinist*, June 2d, 1898, p. 27.

pressed air will eventually shrink on losing the heat of compression. From the theory of air compression,

$$VP_1 = V_1P_3, \text{ or } \frac{V}{V_1} = \frac{P_3}{P_1} \dots\dots\dots\dots\dots (1)$$

$$\text{and } VP_2 = V_2P_4, \text{ or } \frac{V}{V_2} = \frac{P_4}{P_2} \dots\dots\dots\dots\dots (2)$$

But since $P_3 = P_1 + P$, and $P_4 = P_2 + P$, equations (1) and (2) may be written:

$$\frac{V}{V_1} = \frac{P_1 + P}{P_1} = 1 + \frac{P}{P_1} \dots\dots\dots\dots\dots (3)$$

$$\text{and } \frac{V}{V_2} = \frac{P_2 + P}{P_2} = 1 + \frac{P}{P_2} \dots\dots\dots\dots\dots (4)$$

Dividing equation (3) by equation (4):

$$\frac{V_2}{V_1} = \frac{1 + \frac{P}{P_1}}{1 + \frac{P}{P_2}}, \text{ or } V_1 : V_2 :: 1 + \frac{P}{P_2} : 1 + \frac{P}{P_1} \dots\dots \ (5)$$

This gives an expression for the ratio between pressure and volume at sea-level and for any altitude above sea-level, of which the corresponding barometric pressure is P_2. Thus, let $P_2 =$ 10 lbs., $P =$ 90 lbs., and V_1 (from Table VII, page 165) = 0.1404 cu. ft. By substituting these quantities in equation (5), V_2 is found to be 0.0999, or approximately 0.1 cu. ft.

In Table XIII, columns 4 and 5, are given the relative volumetric outputs, at gauge pressures of 70 and 90 lbs. of a compressor working at different altitudes, the figures being percentages of the normal output at sea-level. These percentages have been derived by Mr. Halsey from equation (5), a constant loss of initial pressure of 0.75 lb. being assumed to allow for the resistance presented by the inlet valves, to which reference has been made in another chapter. That is, for practical purposes the sea-level atmospheric pressure is taken as 14, instead of 14.7 lbs. The other columns show the mean effective pressures and indicated horse-powers, corresponding to different altitudes, up to 15,000 ft., which will be found con-

venient for reference. It should be noted from the figures in columns 4 and 5, which are for the ordinary range of pressure employed in mining, that, though there is a difference of 20 lbs. between the two gauge pressures, yet the outputs at different altitudes vary only by a few thousandths and may often be neglected.* Wide differences, however, occur in the columns of mean effective pressures and horse-powers.

TABLE XIII

Altitude, Feet.	Barometric Pressure.		Relative Output for Gauge Pressure.		M. E. P. for Gauge Pressure.		Cubic Feet Piston Displacement per I. H.P. for Gauge Pressure.		Cubic Feet Compressed Air per I. H. P. for Gauge Pressure.	
	Inches Mercury.	Pounds per Square Inch.	70 lbs.	90 lbs.	70 lbs.	90 lbs.	70 lbs.	90 lbs.	70 lbs.	90 lbs.
1	2	3	4	5	6	7	8	9	10	11
0	30.00	14.75	1.000	1.000	33.1	38.2	6.93	5.99	1.144	.801
1,000	28.88	14.20	.967	.966	32.6	37.6	7.03	6.09	1.123	.787
2,000	27.80	13.67	.935	.933	32.1	36.9	7.15	6.20	1.103	.773
3,000	26.76	13.16	.904	.900	31.5	36.3	7.27	6.31	1.084	.759
4,000	25.76	12.67	.873	.869	31.0	35.6	7.39	6.43	1.065	.746
5,000	24.79	12.20	.843	.839	30.5	35.0	7.51	6.55	1.046	.733
6,000	23.86	11.73	.813	.809	30.0	34.3	7.65	6.67	1.028	.720
7,000	22.97	11.30	.785	.780	29.4	33.7	7.80	6.79	1.011	.708
8,000	22.11	10.87	.758	.751	28.9	33.1	7.94	6.92	.994	.695
9,000	21.29	10.46	.731	.723	28.3	32.5	8.09	7.06	.976	.683
10,000	20.49	10.07	.705	.696	27.8	31.8	8.24	7.20	.959	.670
11,000	19.72	9.70	.680	.671	27.4	31.2	8.39	7.34	.942	.658
12,000	18.98	9.34	.656	.647	26.9	30.6	8.54	7.49	.925	.646
13,000	18.27	8.98	.632	.623	26.3	30.0	8.71	7.64	.908	.635
14,000	17.59	8.65	.608	.600	25.8	29.4	8.88	7.80	.891	.624
15,000	16.93	8.32	.585	.576	25.3	28.8	9.06	7.96	.875	.613

Owing to the increase of piston displacement per indicated horse-power, as shown in columns eight and nine of the table, some builders make the air cylinders of compressors for mountain work of larger diameter for the same size of steam cylinder than those for sea-level service. As against the losses of the air end of the compressor at high altitudes, there is some gain in mean effective pressure of the steam cylinders, because the exhaust takes

* Attention may be called to the fact that for this reason, in compressor-builders' catalogues, no account is taken of the gauge pressures in tables of compressor capacities at altitudes.

place against lower atmospheric pressure. The same is true in part of the air exhaust of machines using the compressed air. But the resultant of these gains is small and cannot be given much weight in offsetting the losses. A large deduction, for example, would have to be made for the lower calorific power of a given fuel at high altitudes.

The relation between compressor output and barometric pressure may be expressed simply in another way. Take the case of two compressors of the same size, one operating under an atmospheric pressure of, say, 14 lbs. and the other at 10 lbs. (corresponding approximately to an altitude of 10,000 ft.). If the first compressor is producing 6 compressions, the final absolute pressure will be $14 \times 6 = 84$ lbs. or about 70 lbs. gauge pressure. To produce the same gauge pressure, the other compressor must work to an absolute pressure of $70 + 10 = 80$ lbs., the number of compressions corresponding to which is $\frac{80}{10} = 8$. From each cubic foot of free air the first compressor will produce $\frac{1}{6}$ of a cu. ft. of compressed air, and the second compressor, $\frac{1}{8}$ cu. ft. Hence, the ratio of the respective outputs of the two compressors will be $\frac{1}{8} \div \frac{1}{6} = \frac{3}{4}$ or 0.750. As compared with this, the ratio of the respective barometric pressures is $\frac{10}{14} = 0.714$.

Mechanically Controlled Inlet Valves for High Altitudes. It is often stated that compressors whose inlet valves are under some mechanical control are of special advantage for work at altitudes above sea-level. While there is a measure of truth in this, the possible saving is necessarily small, except at considerable elevations. The question presents itself as follows. If the valve resistance be diminished by introducing mechanical control, so that, under normal conditions at sea-level, the inlet air will begin to enter the cylinder a little earlier in the stroke, the volumetric capacity of the compressor is thereby increased. The loss of capacity due to resistance of the valve springs, etc., which has been assumed to be 0.75 lb., for ordinary poppet valves, is a constant, and therefore becomes proportionately of greater and greater consequence as the altitude increases, because its ratio to the diminishing atmospheric pressure goes on increasing. The

percentage of saving obtained by eliminating the spring resistance, though small at or near sea-level, therefore becomes a matter of importance at great elevations; and the inlet valve which presents the smallest resistance to the entrance of the air into the cylinder will be the most economical for service in high mountain regions.

Stage-Compression at High Altitudes. According to the statement already made, the greater the altitude above sea-level the smaller will be the ratio between the final pressure at delivery and the atmospheric pressure; that is, the ratio of compression. In Chapter V the effect of clearance in the air cylinder was discussed, and it is evident that the percentage loss from this cause increases with the altitude because the piston must advance farther before the clearance air has been re-expanded to a pressure below the diminished atmospheric pressure. Even if it be questioned whether it is worth while at sea-level to adopt stage compression for the ordinary pressures used in mining and tunnelling, the case is materially altered at high altitudes. For example, if it be desired to produce a gauge pressure of 75 lbs. at 5,000 ft. elevation, corresponding to an atmospheric pressure of about 12.2 lbs., 7.15 compressions are necessary. At sea-level this number of compressions would give a gauge pressure of $(14.7 \times 7.15) - 14.7 = 90.4$ lbs. So far as losses due to piston clearance are concerned, therefore, it is as reasonable to employ stage-compression for 75 lbs., at 5,000 ft. elevation, as for 90 lbs. at sea-level. In a compound compressor, too, it must be remembered that there is practically but one clearance space: that in the intake cylinder. The value of the intercooler also increases with the altitude because, in beginning compression at an initial pressure below the normal, the greater total range of pressure through which the air must be carried involves the production of more heat. This additional heat must be effectually dealt with by the cooling arrangements, if loss from this cause is to be avoided.

Considered from both the economic and thermodynamic standpoints, there can be no question as to the value of stage compres-

sion for high altitudes. There is not only a decrease in output and an increase in the cost of production of the air, due to the added power required; but, as a result of these conditions, the compressor itself must be larger for a given output, and therefore its first cost will be greater than that of a compressor of the same capacity, working under normal atmospheric pressure. Hence, by introducing stage compression a larger percentage of saving is possible at high altitudes than at sea-level.

CHAPTER XIV

EXPLOSIONS IN COMPRESSORS AND RECEIVERS

Explosions in air compressors and receivers occur with sufficient frequency to demand careful attention. Though they are unquestionably attributable to ignition of volatile constituents of the lubricating oil, the immediate causes leading to this combustion are not always, nor altogether, clear. It is found, however, that explosions occur only in dry compressors, and some light may be thrown upon the subject by considering the conditions affecting the use of lubricant in these machines. In Chapter V attention was called to the fact that, if the cylinder temperature of a dry compressor be allowed to rise too high, not only does proper lubrication become difficult, but the oil itself may be decomposed by the heat. It is probable that ignition unattended by actual explosion is of frequent occurrence. Instances are on record where the discharge pipe near the compressor has become red-hot, and the ignition even extended into the receiver without producing a destructive explosion. Examination of the discharge-valve chests and passages, and the pipe leading from compressor to receiver, often reveals the presence of a black, sooty residue originating from decomposition of the lubricant. The volatile constituents of the oil thus liberated, on passing with the compressed air into the receiver, would make a mixture of air and gas capable of producing an explosion. The extreme violence often noted in such explosions is probably due in part to the high air pressure existing in the valve passages, discharge pipe, and receiver. In high pressure air, combustion is always more active than in air at atmospheric pressure.

A number of the recorded air-compressor explosions have oc-

curred at collieries, and the possible effects of the presence of coal dust in the intake air of the compressor have been carefully considered. A deposit of such dust in the valve passages, together with the sooty residue from decomposition of the oil, might as a result of oxidation produce a condition very favorable to an explosion. It has been suggested that, in these circumstances, a spark caused by the friction of the compressor piston, if working dry, might bring about an explosion; or, by the continual passage of air at a high temperature over the carbonaceous deposit, spontaneous combustion might result, and ignite the inflammable mixture of oil-vapor and air.* However, there are a sufficient number of cases where explosions have taken place at mines and works other than collieries to prove that such explosions are not necessarily dependent upon the presence of coal-dust in the intake air of the compressor. When the compressor is improperly situated in a room close to the boilers and coal-bins some coal dust might be present in the air; but though possibly assisting in the explosion, the quantity could hardly be large enough to produce by itself the observed results.

The true cause of these explosions is undoubtedly to be found in the working conditions prevailing in the compressor cylinder. In a single-stage dry compressor an excessively high temperature is often reached, because of improper design of the air cylinder, or by running too fast (as when the compressor is too small for its work), or by attempting to produce too high a pressure. The temperature of the discharge air from a single-stage compressor is found by the formula already given in Chapter X:

$$T' = T\left(\frac{P'}{P}\right)^{\frac{n-1}{n} = 0.29}$$

in which: T and P are, respectively, the absolute initial temperature and pressure of the intake air; T′ and P′, the absolute final temperature and pressure; and *n*, the constant, 1.41. Under normal conditions near sea-level, say, when the temperature of the atmosphere is 70° F., P = 14 lbs., and the gauge pressure at dis-

* T. G. Lees, *Trans. Federated Inst. Mining Engineers*, Vol. XIV, p. 568.

charge, 80 lbs., the final temperature is found by making the respective substitutions,

$$\text{whence } T' = 70 + 459^\circ \left(\frac{80 + 14}{14}\right)^{0.29} = 917^\circ \text{ F. absolute,}$$

or 458° F. by the thermometer.

As calculated by this formula, the compression is supposed to be purely adiabatic, no account being taken of loss of heat by radiation or of any effect that may be produced by the water-jackets. As a matter of fact, but little heat can be abstracted by the jackets of a single-stage compressor. Air is a poor conductor, and the volume in the cylinder is not long enough under the influence of the jackets to be much affected by them. In a compressor of this type the chief office of the jackets is to keep down the temperature of the cylinder walls and prevent the lubricating oil from being carbonized. It is probable, therefore, that in a single-stage dry compressor, even if well designed and in good order, the actual temperature of the air at discharge will generally range from, say, 375° to 425° F., and may often go higher—a statement sufficiently supported by recorded observations.

In consideration of what precedes it is evident that the quality of the lubricating oil used in the air cylinder, and especially its flashing- and ignition-points, are matters of importance.* The flashing-point of ordinary cylinder oil may be taken as from 330° to 425° F. "An average of determinations on 40 samples of heavy oils having an average flash-point of 360° F., gave average burning-point of 398° F. High flash test cylinder oils, from 500° to 560° F., gave burning-points of 600° to 630° F." † Common lubricating oils flash at about 250° F., and kerosene, sometimes carelessly used by compressor engineers for cleaning discharge valves, at 150° F. or below. In the case of one explosion the flash-point of the cylinder oil used was found to be only 295° F. ‡

* The flashing-point of oil is the lowest temperature at which it gives off combustible vapors in sufficient quantity to be ignited by contact with flame. The ignition-point is the temperature to which the vapors must be raised in order to continue to burn.

† Alex. M. Gow, *Engineering News*, March 2d, 1905, p. 221.

‡ John Morison, *Trans. North of England Inst. Min. Engs.*, Vol. XXXVIII, p. 6.

It would appear, from a comparison of these temperatures, that an explosion in a compressor cylinder, directly traceable to decomposition of the lubricant, would be possible under normal conditions only when inferior, light mineral oils are employed.

But compressors are not always in good order, nor the working conditions always normal in other respects. Aside from the dangers arising from the use of low-grade lubricant, it is more than probable that one of the commonest causes of explosion is air-cylinder leakage, either of the delivery valves or past the piston. The effects of leakage may be illustrated by citing a case or two.

In 1897 an explosion took place in one of the receivers of the compressor at the Clifton Colliery, England.* It attracted much attention, and is so instructive that many of the details are given here. The air from the compressor passed to a series of 3 receivers of large size, the first being 7 ft. diameter by 40 ft. long. While running apparently under normal conditions the safety-valves of the receivers suddenly began blowing off with a deafening roar. Flames several feet high issued at great pressure from the safety-valves, and sparks were blown out at the joints of the 8-in. pipe leading from the compressor to the first receiver. The air main near this receiver was nearly red-hot. That the receivers did not burst was thought to be due to the relief afforded by the 4 safety-valves—2 on the first receiver and 1 on each of the others—and to the fact that the underground engines driven by compressed air continued running for some minutes after the compressor was stopped. On examining the first receiver, after it had cooled, it was found that, just below the point at which the air entered from the compressor, a mass of black carbonaceous matter had been deposited, from 1½ to 2 ins. thick and 6 sq. ft. in area. On analysis this showed: volatile matter, 55.8 per cent., fixed carbon, 37.3 per cent., and ash, 6.9 per cent. The material was charred and had the appearance of hard vulcanite. A thin coating was noticed on the sides of the receiver (though only near the inlet

* T. G. Lees, *Trans. Federated Inst. Mining Engineers*, Vol. XIV, pp. 555–559.

pipe) and also in the air pipe itself. The other two receivers were free from deposit. A coating of carbonaceous matter, to a thickness of one-quarter inch was found on the discharge valves and passages. The cylinder and piston surfaces were not dry and, though they showed signs of excessive heat, were uninjured.

The gauge pressure was usually 60 lbs., which, with adiabatic compression, would correspond theoretically to a final temperature of 405° F., the temperature of the intake air from the engine-house being 80°. The lubricating oil used was guaranteed to have a flash-point of 554°, and ignition-point of 606° F. As the cylinders were water-jacketed, the actual final temperature should not, in regular working, reach the abovenamed temperatures; in fact, readings previously taken from a thermometer in the outlet pipe showed that it usually registered about 350° F. It is significant, however, that on one occasion the mercury rose above 500°, and the thermometer tube burst. The temperature at the time of the explosion therefore was not known. Afterward a pyrometer was fixed on the outlet pipe as near as possible to the discharge valves, and the temperature was found to range generally from 400° to 420° F., varying with the speed of the engine and the air pressure produced. Even with these temperatures, high as they are, it would seem impossible that ignition of the lubricating oil could take place. It is evident that an unusual increase of temperature in the air cylinders must be accounted for.

In commenting on this accident, Mr. W. L. Saunders makes the following interesting remarks on explosions in compressors and receivers:

"There must be an increase of temperature, or ignition would not take place. This increase of temperature may result either from an increase of pressure, which is not recorded on the gauge, or there may be an increase of temperature without a corresponding increase of pressure. Take the first instance, and it is not difficult to understand that a compressor might deposit carbon from the oil in the discharge passages or discharge pipes, which in the course of time will accumulate and constrict the passages so that

they do not freely pass the volume of air delivered by the compressor. Hence, a momentary increase of pressure might exist in the cylinder heads, or in the discharge pipe which leads from the cylinder to the receiver, which would surely carry with it an increase of temperature possibly exceeding the ignition-point of the oil. A badly designed compressor with inefficient discharge passages might produce this trouble. Too small a discharge pipe or too many angles in discharge pipes might also tend to produce explosions. But ignition is known to have occurred in a well-designed system, and other causes must be sought. We think many cases may be traced to an increase of temperature without increase of pressure; this increase of temperature can be excessive only when the temperature of the incoming air is excessive. A hot engine-room from which air is drawn into the cylinder is a bad condition. Ignition is known to have taken place, however, when the temperature of the incoming air was normal, when the discharge passages and pipes were free and of ample area, so that some other cause must still be looked for. The only possible explanation is that the temperature of the intake air is made excessive by the sticking of one or more of the discharge valves, thus letting some of the hot compressed air back into the cylinder to influence the temperature before compression. . . . It is not difficult to understand a leaky discharge valve letting enough hot compressed air back into the cylinder to increase the initial temperature to 200 or 300°. If so, and the air is being compressed to 73.5 lbs. gauge pressure we have, say, 300° temperature in the free air before compression, and as the increase is 354.5°, the resulting temperature might be 654.5°. As a remedy we would suggest more care in selecting the best compressor, and in frequent cleaning of the discharge valves and passages. The best compressors are built so that the discharge valves may be readily removed. These valves should be cleaned once a week by the engineer, who should see that they fit properly. It is impossible to get good lubricating oil that is free from carbon, hence there will always be more or less carbon deposited on the discharge valves, but this must not be allowed to accumulate.

Intercoolers between air cylinders and aftercoolers between final cylinder and receiver are also recommended. One of these coolers located in the discharge pipe will absolutely prevent the passage of flame, and will insure the protection of the mine against fire even though there be ignition at or near the air cylinder." *

During the construction of the New York Aqueduct a fire occurred in a compressor receiver at one of the shafts. The air pressure was eighty to ninety pounds, and the horizontal receiver, set outside of the engine-house, was exposed to the hot sun. Part of the discharge pipe leading to the receiver had become red-hot. On stopping the compressor and cooling down the receiver, the entire inner surface of the latter was found to be coated with carbonaceous matter at least one-eighth inch thick. Further investigation brought out the fact that the poppet discharge valves had sometimes occasioned trouble by sticking, and the engineer had been in the habit of using a squirt-can of kerosene to cut the gummy material clogging them. As the kerosene had a low flash-point, it was quickly vaporized, and when the cylinder temperature reached a sufficiently high point the explosion took place.

In this case, as in that previously cited, the trouble seems to have been caused by leakage of the delivery valves (possibly past the piston also), thereby raising the cylinder temperature to an abnormal degree. It may be added that the use of kerosene for cleaning gummy discharge valves is a dangerous practice, even when the compressor is slowed down while using it.

The effect of leakage in the air cylinder may readily be understood from the following discussion of what takes place in the course of a single stroke, with the accompanying temperature changes.† At the beginning of the stroke, the air in the cylinder consists of: that which remained in the clearance spaces at the end of the previous stroke, that which has leaked in, and that which has been drawn in from the atmosphere. The clearance air, on

* *Compressed Air*, July, 1897, pp. 258–259.

† Abstracted from a paper by E. Hill, *Trans. Amer. Inst. of Min. Engs*, Vol. XXXIV, p. 950.

re-expanding, falls from an absolute temperature of T′ to T (see formula near the beginning of this chapter), and its effect may therefore be neglected. For well-designed compressors, the temperature of the air newly drawn into the cylinder may be taken as that of the outside atmosphere, t, though it is generally heated in some degree by contact with the hot inner surfaces of the cylinder. Finally, if L represent the volume of air leakage, then, since T is the absolute temperature of the entire mass of air occupying the cylinder at the beginning of the stroke:

$$T=(1-L)\,t+T\,L \ldots\ldots\ldots\ldots (1)$$

If, in the expression previously given for the temperature of the discharge air, *viz:*

$$T'=T\left(\frac{P'}{P}\right)^{0.29}$$

the pressures be written in atmospheres; then, for compressors working at sea-level, P = 1 and:

$$T'=T\,P'^{\,0.29},\ \text{whence}\ T=\frac{T'}{P'^{\,0.29}} \ldots\ldots\ldots (2)$$

Placing the values of T, in equations (1) and (2) equal to each other and transposing:

$$T'=\frac{t\,(P^{0.29}-L\,P'^{\,0.29})}{1-L\,P'^{\,0.29}}$$

Applying this formula to a single-stage compressor, working to say, 7 atmospheres, or about 88 lbs. gauge, the atmospheric air being at 62° F., the discharge temperatures for different percentages of leakage will be as shown in Table XIV. The temperatures for an altitude of 4,000 ft. are also given for purposes of comparison. The leakages are expressed as percentages of cylinder capacity.

These possible temperatures are fully sufficient to produce an evolution of gas, or even decomposition of the cylinder oil, causing it to burn; which would be followed by an increased liberation of volatile matter and the probability of explosion.

It must be borne in mind, as pointed out by Mr. E. Hill, that leakage from an imperfectly fitting discharge valve is a constant in any given case, while the volume of intake air varies with

the speed of the compressor. Thus, a leak of 2 per cent. of the intake volume, at, say, 125 revolutions per minute, becomes 10 per cent. if the compressor be slowed down to 25 revolutions. This agrees with experience, violent explosions being known to have occurred while the compressor was running slowly. "The oil-feed was probably adjusted to the maximum speed and hence was excessive for the slow speed. A larger proportional leak—a liberal quantity of oil—and the result is easily comprehended."

TABLE XIV

Leakage. Per Cent.	Temperature of Discharge. Degrees Fahrenheit.	
	At Sea-Level.	At 4,000 Feet Elevation.
0	459	496
1	466	504
2	475	513
4	489	530
6	506	549
8	524	570
10	544	593
12	566	618
14	589	646
16	615	675

The effect of leakage of the discharge valves, moreover, is cumulative, for each rise in initial temperature thereby produced causes a greater rise in terminal pressure; and the leakage continuing, a very few strokes would suffice to ignite the best cylinder oil. In some circumstances, even a single stroke of the piston may cause ignition, if not explosion.

The importance of minimizing piston and discharge-valve leakage is evident. One of the surest means of avoiding danger of high cylinder or receiver temperatures is the adoption of stage compression. There are two reasons for this: (1) the air is partly cooled between the stages, so that the maximum temperature is always less than in single-stage machines, and (2) the leakage is likely to be less because there is a smaller difference between the pressures on the two sides of the piston, as well as between the internal and external pressures on the discharge valves.

A case of explosion, in which the influence of cylinder leakage is not clearly apparent, occurred some years ago in the air pipe of a large plant in Butte, Mont. Two duplex compressors, with air cylinders respectively of $32\frac{1}{4} \times 60$ ins. and $24\frac{1}{4} \times 48$ ins., and running at 50 revolutions per minute, were forcing air at 80 lbs. pressure through a single 8-in. pipe. As somewhat over 1,200 cu. ft. of compressed air per minute were being produced, the velocity of flow would be nearly 3,500 ft. per minute, or 58 ft. per second. It had been noticed several times that a portion of the discharge pipe close to the compressor became red-hot. As no explosion took place in the compressor cylinders, but in the pipe only, it is probable that the oil accumulated in the pipe was vaporized and ignited. In the pipe between compressors and receivers there were several sharp bends, which increased the friction due to the rapid flow of the air. The receivers were always extremely hot. On one occasion the shaft timbering, forty or fifty feet below the shaft mouth, took fire from the hot air pipe. The above gives point to the fact that, while the primary causes of explosion are to be found in the air cylinder, the disastrous effects are perhaps oftener observable in the discharge pipe or receiver.

Foul or poisonous gases may result from ignition of the lubricant in compressors or receivers, not necessarily followed by actual explosion. In an article in the *Trans. Amer. Inst. Min. Engs.*, Vol. XXXIV, p. 158, an instance is noted of combustion in the air pipe and receiver. The compressed air was being used in an imperfectly ventilated upraise in the mine, 1,200 ft. from the compressor, and 2 men lost their lives, while 4 others barely escaped asphyxiation.

Other more or less similar cases are familiar to most miners, where foul air from the exhaust of machine drills has been observed; sometimes merely disagreeable, though often actively deleterious. The use of poor cylinder oil is frequently responsible for this, as its lighter constituents may begin to volatilize and burn at a perfectly normal working temperature. Even if not actually fried on the hot metal surfaces, a low-grade oil may yet undergo

a slow combustion or oxidation, which will produce enough carbon dioxide to raise materially the percentage of that poisonous gas in the confined atmosphere of the working places of mines.

The mode of using the lubricant for the air cylinders of compressors deserves some attention. Sight-feed lubricators, such as are commonly employed for steam cylinders, are best. On the Clifton Colliery compressor, mentioned above, ordinary oil-cups were used, holding about ⅛ pint. They were filled 4 times per day of 10 hours. With these oil-cups, if improperly adjusted, it would be possible for all the oil to be sucked into the cylinder within a few strokes after being filled. Such a result might be inferred, indeed, in this case, because of the large quantity of carbonaceous matter—oil, coal dust, etc.—found in and around the discharge valves and in the receiver. The feeding of the oil should be carefully regulated, and a smaller quantity used in an air cylinder than a steam cylinder of the same size—say, one-third as much. An excess of oil increases the tendency to gum the valves. For stage compressors of ordinary size, 1 drop of good cylinder oil every 4 to 5 minutes is sufficient.

The periodical use of soap and water (soap-suds) is to be recommended for any compressor that cannot be shut down at short intervals for overhauling. It is fed into the air cylinder through an oil-cup, say during one day per week. Or it may be forced in by an oil-pump, with which the air cylinder should be provided. Soap and water is a poor lubricant in itself, and must be used more freely than oil, but it is effectual in cleansing the cylinder, valves, and ports from any carbonaceous or gummy matter that may have been deposited. If the compressor is to be stopped, as at the end of a shift, care must be taken to discontinue the feeding of soap and water some time before shutting down, and resume the oil-feed. This is necessary to avoid the formation of rust. Every compressor should be overhauled from time to time, and thorough cleaning should extend to all parts, especially around the valves and passages, capable of furnishing a lodgment for oil or partly oxidized carbonaceous material.

Precautions for Preventing Explosions. These may be sum-

marized as follows: (1) Always enclose the inlet valves in a cold-air box, connecting with the outside air, so as to avoid taking the air from the hot engine-room. This not only conduces to economy in working, but by keeping down the final temperature tends to prevent decomposition of the oil. (2) The largest possible area of cylinder surface should be water-jacketed, including the cylinder heads. A liberal supply of the coldest water obtainable should be used for the jackets. The advantages in this respect derived from employing stage compression, with large inter- and aftercoolers, are undoubted. (3) Use only the best cylinder oil, with high flash- and ignition-points and in as small quantity as is consistent with proper lubrication. Care should always be taken to keep the valves clean. In the design of the compressor there should be no recesses or pockets, around the valves or passages, where oil could accumulate. (4) So arrange the air intake that coal dust will not be drawn into the cylinder with the inlet air. (5) It is well to place a thermometer in the discharge pipe, close to the cylinder, so that the engineer will be able to note the temperature from time to time and stop or slow down the compressor if the temperature of the discharge air rises too high.

CHAPTER XV

AIR COMPRESSION BY THE DIRECT ACTION OF FALLING WATER

In view of the economic importance of keeping down the temperature of the air during compression, it is evident that an advantage would be derived from a closer and more intimate contact between the air under compression and the cooling water than is possible with the external water-jackets of dry compressors. From a thermodynamic standpoint it cannot be questioned that the wet compressor is more efficient than the dry. As has been shown in the latter part of Chapter V, it is mainly the mechanical difficulties resulting from the use of injected water in the air cylinder that operate to the disadvantage of the wet system of compression, and that have caused its almost complete abandonment.

Since 1896 several large plants have been successfully installed in which air is compressed by the direct action of falling water and without the use of piston, valves or other moving parts. The simple and familiar principle involved has aroused much interest in this method of air compression. When air in small bubbles is intimately mixed with water, the water breaks into foam, through which the air bubbles tend to rise and escape. But if the mixed air and water be drawn downward by a strong falling current, suitably confined, as in a vertical pipe, the air is compressed. And if, after reaching the depth and head of water column necessary to produce the degree of compression desired, the direction of flow be changed to the horizontal and the velocity diminished, the air bubbles will rise. They may then be collected in a suitable chamber, in which the air pressure corresponds to the head of water and from which the air is drawn off as required.

As the air bubbles are minute and thoroughly disseminated through the water during its descent, the total cooling surface presented is very large and complete isothermal compression results. It should be observed also that the compressed air is very dry. While undergoing reduction in volume, the percentage of moisture in a given globule of air increases until the point of saturation is reached, but any further compression causes deposition of part of the moisture. Moreover, since the air is kept constantly cool during compression, its moisture-carrying capacity is smaller than if compressed adiabatically, as in an ordinary compressor cylinder.

Although such an apparatus embodies no new principle, it was first constructed on a working scale and successfully tested, about 1878, by J. P. Frizell, of Boston, Mass.* Aided by this precedent, a more effective and practical method of breaking up the water and impregnating it with air in a state of fine division, was afterward devised by Charles H. Taylor, of Montreal, Canada. In 1896 the Taylor Hydraulic Air Compressing Co., of Montreal, erected a plant embodying the system for the Dominion Cotton Mills, Magog, Province of Quebec.† This plant has long been in successful operation, and where the conditions permit its introduction the system may be advantageously employed for mining service also.

For the Magog Mills a 128-ft. shaft was sunk to give the desired head and pressure (Fig. 117). In it was erected a large vertical compressing pipe, *a*, 3 ft. 8½ in. diameter, the lower part gradually increasing to 4 ft. 8 in., and made of $\frac{5}{16}$-in. steel plate. This pipe passes through the bottom of an iron receiving chamber, *b*, at the surface, to which water is conducted from a dam or reservoir. The chamber, *b*, is 12 ft. diameter by 12 ft. high. Water flows into and fills the pipe, which extends nearly to the

* For a record of these tests see *Proceedings of the Institution of Civil Engineers*, London, Vol. LXIII, p. 347.

† The following description is based on an article in the *Canadian Engineer*, March, 1897, and information furnished to the author by the builders. See also *Eng. and Mining Jour.*, Dec. 26th, 1896, p. 606, and *Railway and Engineering Review*, Sept. 17th, 1898, p. 513.

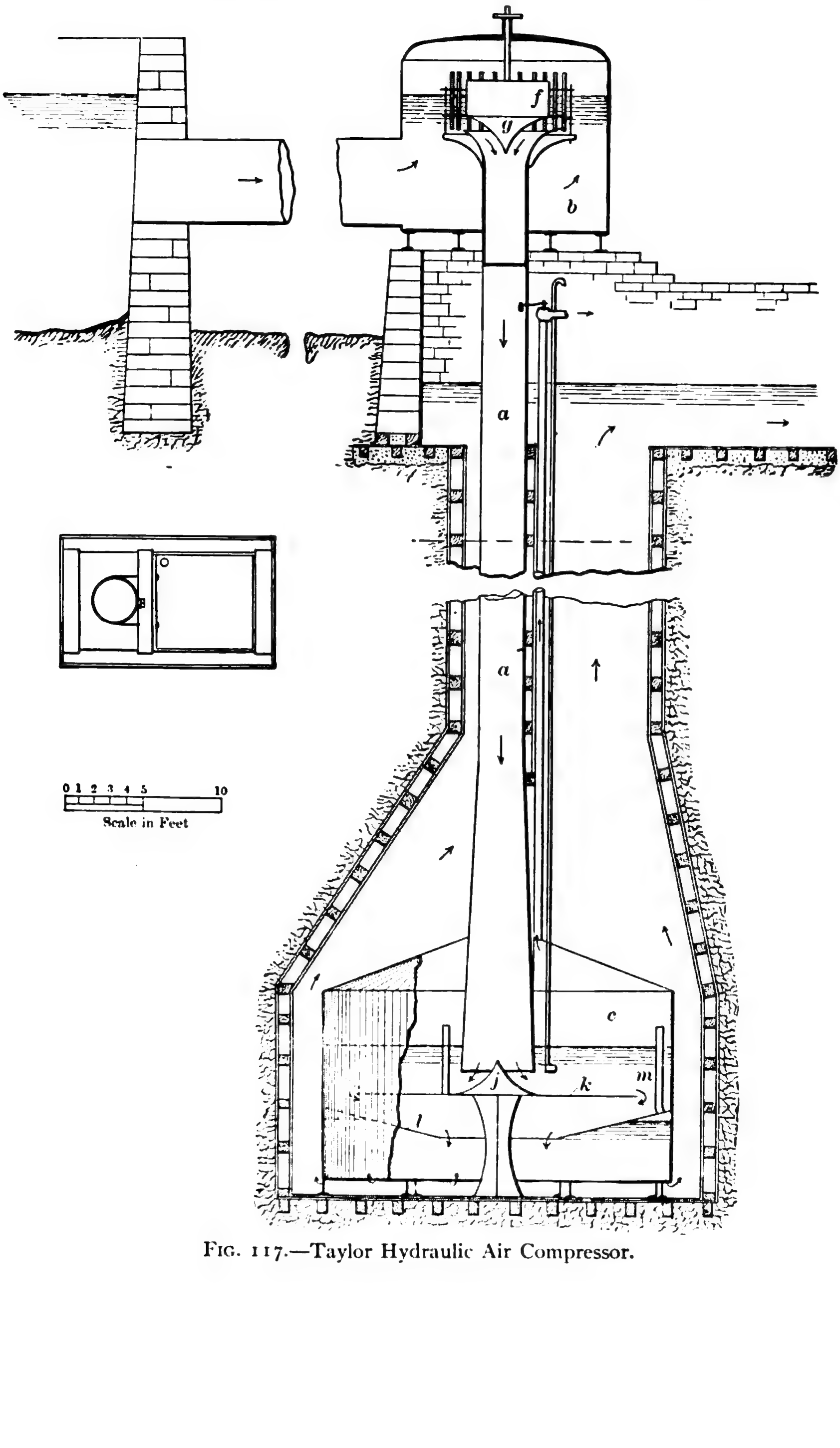


FIG. 117.—Taylor Hydraulic Air Compressor.

bottom of the shaft. By means of an arrangement of small feed pipes described below, air is drawn with the water into the top of the main vertical pipe and is compressed while being carried down the shaft. The compressed air collects in a separating

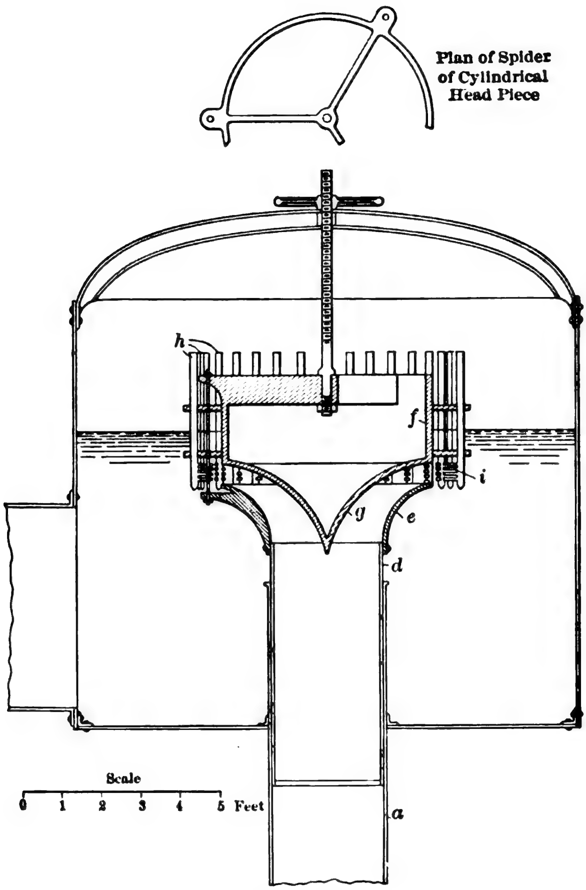


FIG. 118.

chamber, *c*, at the bottom of the shaft, while the water is returned up the shaft to a tailrace near the top. The difference of water level between intake and tailrace is about 22 ft., which produces the requisite speed of flow of the mass of water. Into the top

of the vertical pipe, *a*, is inserted a telescoping section of pipe, *d*, to the upper end of which is riveted a bell-mouth, *e*. Above the latter is a cylindrical headpiece, *f*, 4 ft. 8 ins. diameter (Fig. 118), terminating below in an inverted conoid, *g*, projecting into the bell-mouth. These two parts are connected by lugs and bolts in such way as to leave an annular opening between them, through which the water enters the vertical pipe. Around the headpiece is set a series of thirty 2-in. pipes, *h*, *h*, 4 ft. long, open at the top and closed at the bottom. Into each of these pipes, near their lower ends, are screwed 32 short horizontal $\frac{3}{8}$-in. pipes, *i*, *i*, all directed into the annular opening at the bell-mouth and toward the axis of the main pipe. As the entering water passes among the small pipes a tendency to vacuum is created in them, so that the atmospheric pressure drives the air through them into the water in the form of small bubbles. These are carried with the water down the main pipe, and on their way are compressed.

Near the bottom of the shaft the vertical compressing pipe enters the large circular "separating" chamber, *c*, 17 ft. diameter and 12 ft. high, open below and supported upon legs which raise it 16 ins. above the shaft bottom. Within the tank and directly under the pipe is the "disperser," *j*, a conoidal casting like the one in the headpiece. Plates, *k*, are added around the periphery of the disperser to give it an outside diameter of 12 ft. Below is an inverted conical apron, *l*, 5 ft. wide, riveted to the interior of the separating tank. When the water, charged with air bubbles, reaches the disperser it is directed outward toward the circumference; is then deflected by the apron toward the center under the disperser, and finally escapes through the open bottom of the separating tank into the return column. During this process of travel the compressed air separates from the water, most of it collecting in the upper part of the air chamber. A portion of the air is not liberated until the water reaches the lower part of the tank, under the apron. This residuum collects in the annular space and joins the main body of air through the pipe, *m*. The compressed air collecting in the top of the air chamber is kept under pressure by the weight of the re-

turn water column in the shaft, and is drawn off through the vertical air main, alongside of the water column *a*. As the small air bubbles are constantly surrounded by cold water, it is evident that by this system perfect isothermal compression is attained, with its corresponding advantages in minimizing the amount of moisture carried off in the air. This has been shown by tests.

With a total depth of shaft of 128 ft., in this installation, an air pressure of 52 lbs. per sq. in. is produced. The efficiency of this plant is shown by the following table* to be from 50.1 per cent. to 62.4 per cent., according to the quantity of water used:

TABLE XV

No. of Test.	Quantity of Water Discharged, in Cubic Feet per Minute.	Available Head in Feet.	Available Horse-Power.	Quantity of Air Delivered, in Cubic Feet per Minute at Atmospheric Pressure.	Pressure of Air, Pounds per Square Inch.	Actual Horse-Power of Compressor.	Efficiency of Compressor, per Cent.
1	6122	21.4	247.7	1377	52	132.5	53.5
2	5504	21.9	228.0	1363	52	131.0	57.5
3	4005	22.3	168.9	1095	52	105.3	62.4
4	7662	21.1	305.9	1616	52	155.4	50.8
5	6312	21.7	260.0	1506	52	144.8	55.7
6	7494	21.2	299.8	1560	52	150.2	50.1

Temperatures during tests: external air 75° to 83°; water 75.2° to 80°; compressed air 75.2° to 80°.

The parts were not correctly proportioned in this first installation, and there is no doubt that the efficiency could be considerably increased by using a relatively larger air chamber at the bottom of the shaft, to prevent air from going to waste. As shown by the table, the efficiency is increased by diminishing the volume of inlet water, upon which depends the quantity of air carried down and compressed.

In building a plant to produce higher air pressure the motive head, or difference in level between the surfaces of water at inlet and tailrace, would be increased. The theory is as follows: The

* Tests made by Prof. C. H. McLeod, of McGill University, August, 1896. Published in *Eng. and Min. Journal*, December 26th, 1896, p. 606.

combined specific gravity of the mixture of air and water in the vertical compressing pipe is less than that of the water in the return column. That is, the weight of water in the compressing pipe is less per foot than in the return column. Therefore, the head required, to overcome friction and to produce flow, must be greater than if the apparatus were merely an inverted siphon, and as the difference in weight increases with depth (and air pressure produced) the motive head must be correspondingly increased.

In 1898–1900 a plant on the Taylor system was built for the Kootenay Air Supply Co., Ainsworth, British Columbia. The topographical conditions are such that a high head of water is obtained without sinking a deep shaft. From a small dam the water is carried in a wooden-stave pipe, 5 ft. in diameter and 1,354 ft. long. The pipe finally passes over a short, but high trestle, built against the side of a steep gorge, to the receiving tank. The latter, 17 ft. diameter by 20 ft. high, is placed on a wooden tower, 110 ft. high (Fig. 119). From the bottom of the tank the pressure pipe, 33 ins. diameter, descends vertically inside the tower to the ground level and then down a shaft 105 ft. deep.* After compressing the air the water returns up the shaft to the tailrace at the creek level. As shown in Fig. 120, the details of the receiving chamber at the bottom of the shaft differ from those of the Magog plant.

The effective compressing head is 107 ft., while the total height of the pressure pipe is over 200 ft. This produces a high velocity of flow and a correspondingly large delivery of compressed air. The main pipe line, 9 inches diameter, is 2 miles long, discharging from 4,200 to 4,600 cu. ft. of free air per minute. Branch service pipes convey the air to neighboring mines, where it is used for rock-drills and other mining machinery. On the basis of 600 horse-power, represented by the volume and pressure of the air compressed, the cost of the entire plant, including pipe lines, was about $100 per horse-power. This would be somewhat increased by allowances for transmission and other losses.

* *Canadian Electrical News*, September, 1898, p. 176.

Another large plant was completed in 1906 at the Victoria Copper Mine, near Rockland, Ontonagon Co., Michigan. Though the same general design was adopted for the intake head

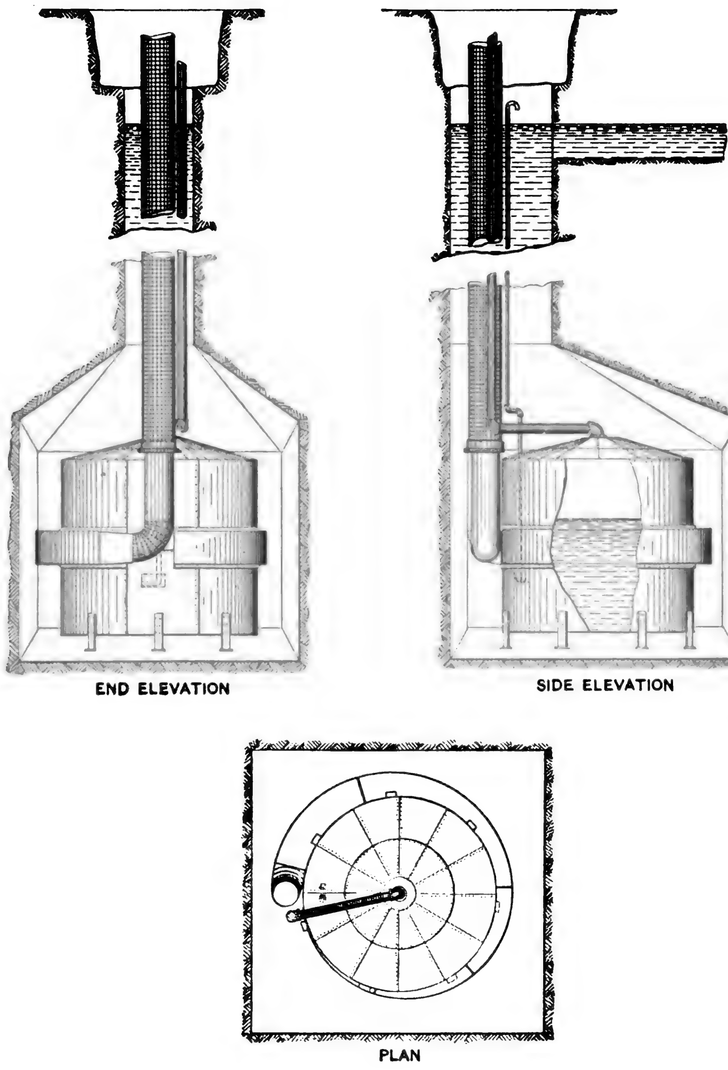


FIG. 120.—Hydraulic Air-Compressor at Kootenay.

and its appurtenances, the local conditions led to a novel mode of installation. The water is conducted from a dam on the Ontonagon River through a 4,700-ft. canal, furnishing a head at the terminal forebay of 72 ft. above the river-level. Three independent units are built side by side in a vertical shaft 343 ft. deep. The subdivision of the air, as admitted at the intake head, is carried farther than in either of the plants described above, there being no less than 1,800 $\frac{3}{8}$-inch horizontal feed pipes, inserted in the series of vertical pipes encircling the inverted cone. The compressing pipes are 5 ft. in diameter, lined with concrete, and the separating cones and dispersers, also of iron and concrete, are built at the bottom in a rock chamber excavated for the purpose. In this chamber, 281 ft. long and 18 ft. $\times$ 21 ft. average cross-section, the compressed air is trapped and thence drawn off for use through a 24-inch main. The compressing water, flowing down the intake pipes, stands normally at a level about 14$\frac{1}{2}$ ft. below the roof of the chamber, thus leaving an air capacity of about 80,000 cu. ft. Connected with the end of the air chamber is an inclined shaft, 270 ft. in vertical depth, through which the water returns to the surface. The tailrace from the mouth of this shaft is 72 ft. below the level of the intake, this height measuring the motive head producing the flow of water. Thus the air in the underground chamber is under a pressure due to 270 ft. head of water, or 117 lbs. per sq. in. gauge.

For regulating the operation of the compressor a pipe passes from the air chamber up the compressing shaft to the surface, where branches from it are led to the intake heads. The compressed air conveyed in this regulating pipe operates a device connected with each intake head, whereby the latter is automatically raised above the water-level in the receiving tanks whenever the air pressure exceeds the normal, thus stopping the flow of air through the feed pipes. A twelve-inch blow-off pipe is also provided, passing from the water-level in the air chamber to the mouth of the inclined shaft carrying the return water column. If air to the full compressor capacity is drawn off, the water-level in the air

chamber rises as the air pressure falls, thus sealing the lower end of the blow-off pipe; then, when the consumption of air decreases the pressure in the chamber rises, depressing the water-level until the blow-off orifice is uncovered, when more air is blown off. Thus the working pressure is maintained within quite narrow limits. It may be added that the great size of the air chamber—which acts like the receiver of an ordinary air-compressor plant—gives it a large storage capacity.

When all 3 compressing units are in operation, with a total capacity of from 34,000 to 36,000 cu. ft. of air per minute, about 70,000 cu. ft. of free air per minute may be drawn off for a period of 18 minutes, without causing a drop in pressure of more than 5 lbs. For each unit, the output ranges from 9,000 to 12,000 cu. ft. per minute, and the volume of water used, from 12,700 to 14,800 cu. ft. A series of tests made on a single intake head in May, 1906, by Prof. F. W. Sperr, gave the following results: *

TABLE XVI

AIR MEASUREMENTS

Square Feet.	Velocity, Feet per Second.	Cubic Feet per Minute.	ABSOLUTE PRESSURES		Horse-Power.
			Free Air, Pounds.	Compressed Air, Pounds.	
4	44.09	10,580	14	128	1,430
4	49.74	11,930	14	128	1,623
4	38.50	9,238	14	128	1,248

WATER MEASUREMENTS

Flume Area.	Velocity, Feet per Second.	Cubic Feet per Minute.	Head, Feet.	Horse-Power.	Efficiency, per Cent.
71.75	3.033	13,057	70.5	1,741	82.17
67.03	3.684	14,820	70.0	1,961	82.27
72.16	2.936	12,710	70.6	1,700	73.50

The air is used at the Victoria Mine for general power purposes at the mine and mill, including a 500-horse-power hoisting

* For further details see article by D. E. Woodbridge, *Engineering and Mining Journal*, Jan. 19th, 1907, p. 125. Also, A. H. Rose, *Mines and Minerals*, March, 1907, p. 346.

engine, designed for a depth of 4,000 ft., 7 pumps, and many other engines. The cost per horse-power is only about $2.25 per year, including all the operating expenses. It is expected that over 4,000 horse-power will be developed when all 3 compressing units are in operation. The present stage of the development of the mine requires the use of but 1 unit.*

The compression of air by direct action of falling water, according to the Taylor system, has been adopted in several other recent installations: two in Germany and a very large plant for general power purposes, on the Shetucket River, near Norwich, Conn.† It is probable that the application of the system will be extended in regions where large water powers can be developed. Its first cost is not excessive, while the maintenance and running expenses are extremely low, as compared with those of the usual forms of air compressors. No skilled attendance is required, and the item of depreciation is merely nominal in such substantially erected plants as that at the Victoria Mine. By comparing the figures given in Tables XV and XVI, it will be seen that in the later installation a very marked increase was made in efficiency of operation; due to improved design of the intake head, increase in motive head producing the flow of the compressing water, and a more complete separation of the air from the water in the receiving chamber.

It has been suggested that it might be feasible to employ the system in connection with an ordinary compressor plant. That is, to produce a low air pressure by the water plant, and then to admit this air to the compressor cylinder where it would be brought up to the required higher tension. In effect, this would be stage compression, in which the air would be completely

* Since the Victoria plant was put in operation, trouble has been experienced by the freezing up of the small pipes of the intake heads, due to the severe winter climate of the region. This has led to the removal of the heads, as originally designed, the water being allowed simply to flow into the top of the compressing pipes. I am informed that the capacity of the plant, in cubic feet of free air compressed per minute, is practically the same as when the intake heads were in use (May, 1909).

† The last-mentioned is described in *Compressed Air*, April, 1906, p. 3,980.

cooled to normal temperature before entering the high-pressure cylinder.

In 1907–8 an underground hydraulic air compressor was installed at one of the silver mines of Clausthal, Germany. Fig. 121 shows, in plan and elevation, the general design of the plant, with details of the intake head and compressing chamber. A flow of water in the tunnel *t* is led through an 8½-inch cast-iron pipe, *a*, to the vertical air intake, *b*. This, shown in longitudinal section in the detail cut, consists of a number of flaring cast-iron rings, *l*, in the upper rim of each of which is a series of small holes, *k*, for admitting the air. Additional inlet area is provided at the top of the intake by a nest of small curved pipes, *m*. The air is thus drawn into the pipe and entrained by the downward flow of the water. The mixed air and water pass into the 8½-inch compression pipe, *c*, 492 ft. long. This pipe is laid in an inclined shaft, and discharges into the bottom of the compressing chamber, *d*, shown also in detail. The chamber is 52 in. diameter by 14 ft. 9 in. high. From a point near its top the compressed air passes through the pipe *n* to the automatic check-valve *e*, and thence, by the pipe *h*, to the receiver *i*. Pipe *p* conveys the air from the receiver to the working places of the mine. The water leaves the compressing chamber by the 8½-inch pipe *f*, which discharges at a point 164 ft. above, into a tail-race occupying the mine level *u*. An equalizing discharge pipe, *g*, from the compressing chamber, is led up the shaft, parallel to *f*, entering the latter at the level of the tail-race. The total cost of the plant, installed, is stated to be $3,750.*

The average flow of water is 792 gallons per minute; which, falling through a vertical height of 325 ft. (from intake to discharge at the tail-race), produces theoretically 66.3 horse-power. In testing the plant, the water was measured by a weir and the quantity of air compressed by filling a receiver of known capacity. It was found that a flow of 845 gallons per minute gave 353 cu. ft. of air, at a gauge pressure of 71.2 lbs. To compress 1 cu. ft. of

* Abstracted from a description by P. Bernstein, in *Glückauf*, March 14, 1908. Translation by E. K. Judd in *Engineering and Mining Journal*, August 1, 1908, p. 228.

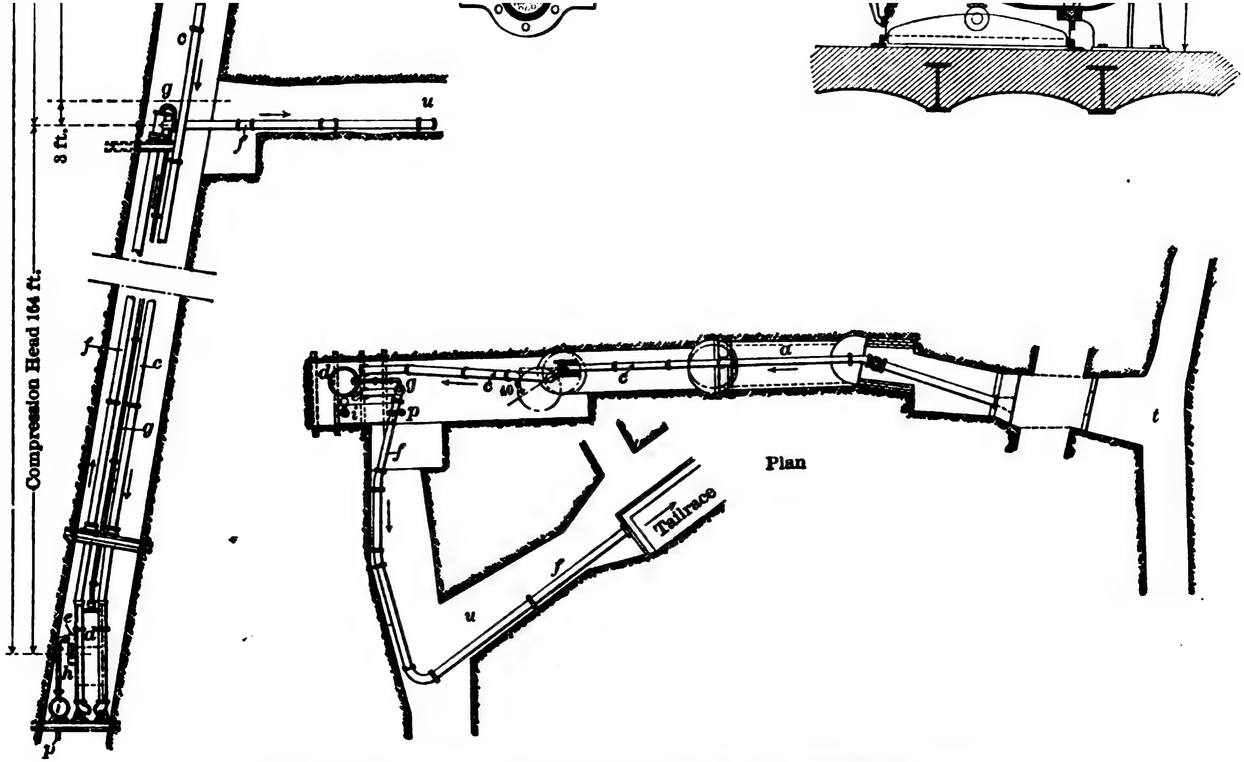
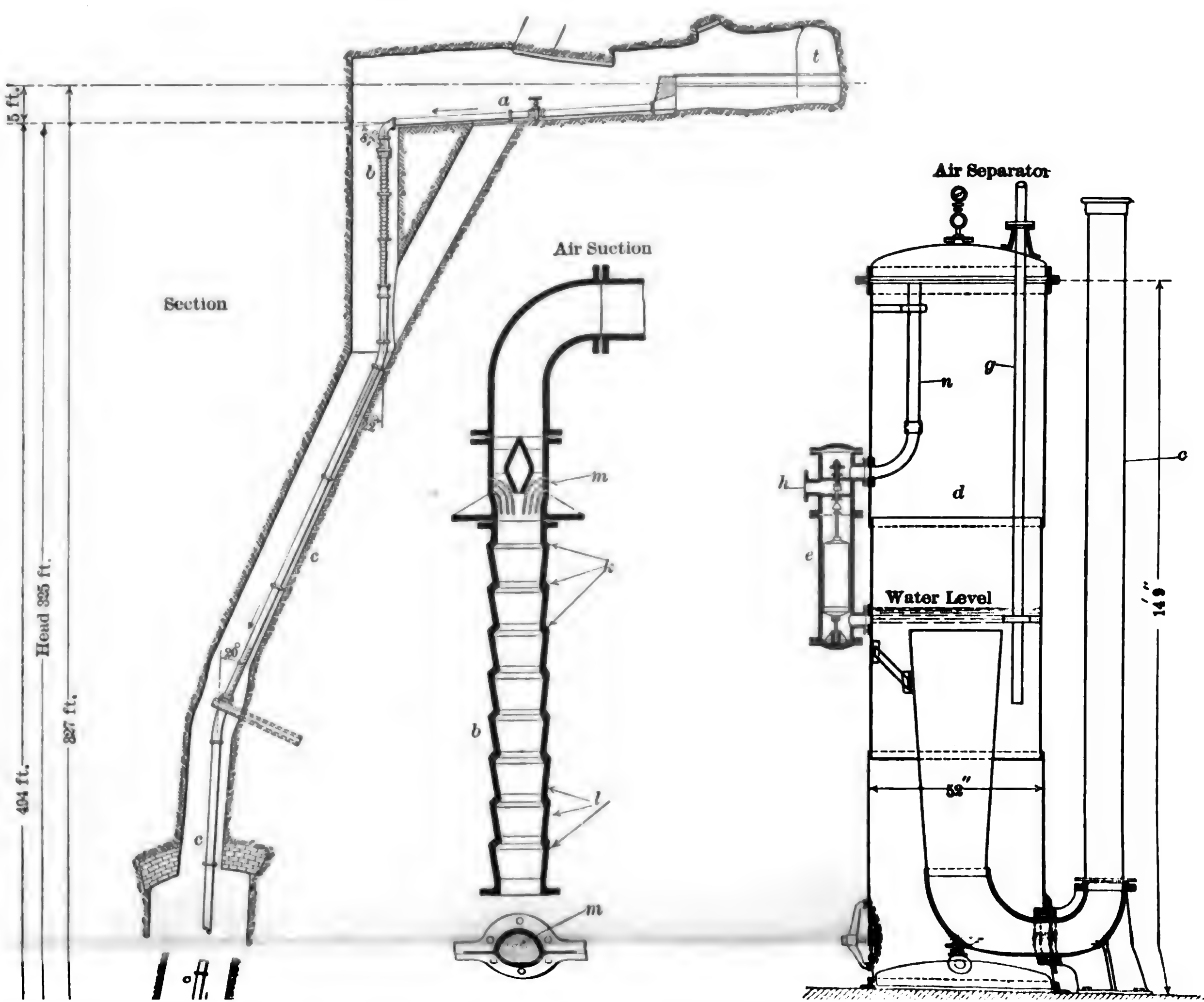


FIG. 121.—Underground Hydraulic Air Compressor, at Clausthal, Germany.


5 ft.
Head 325 ft.
494 ft.
827 ft.
Section
Air Suction
Air Separator
Water Level
52″
14′ 9″
a
b
c
d
e
g
h
l
m
n
t

air adiabatically to this pressure requires 0.147 horse-power and to compress 353 cu. ft., about 51.9 horse-power. Since 70.5 theoretical horse-power is produced by the flow of 845 gallons per minute, the efficiency is $\frac{51.9}{70.5} = 73.6$ per cent.

Part Second

TRANSMISSION AND USE OF COMPRESSED AIR

CHAPTER XVI

CONVEYANCE OF COMPRESSED AIR IN PIPES

CERTAIN losses due to friction take place in conveying compressed air through lines of piping. The diameter of the pipe is of vital importance, and when proportioned properly to the volume of air, and to the distance, these transmission losses are very small as compared with the other losses incident upon air compression. With the possible exception of electricity, no other means of power transmission can compare in efficiency with compressed air. The transmission losses appear in two ways: as loss of power, and as loss of pressure or head, indicated by difference in gauge reading at the ends of the line. Between these two losses there is a clear distinction.

Loss of Power. The large and, to a great extent, unavoidable loss of power due to the heating of the air during compression and its subsequent cooling after leaving the compressor, has already been considered. But this cooling takes place so quickly in the receiver and piping that the resulting loss is not properly chargeable to transmission. The air assumes the temperature of the surrounding atmosphere in the first few hundred feet, so that when conveyed to long distances the calculation for transmission loss may be made without regard to the effect of temperature upon the volume of the air. In other words, the volume is taken simply as proportional to the absolute temperature, in atmospheres.

The power residing in the compressed air is due not only to its pressure, but also to its volume, in terms of number of cubic feet of free air (*i.e.*, air at atmospheric pressure). Thus, while the pressure is reduced by frictional loss in transmission, yet this reduction in pressure is accompanied by a proportionate increase in volume, and a certain compensation is produced. Although the pressure of the air at the motor is diminished, there is no loss in the final volume of free air. As will be shown below, the loss of pressure due to the conveyance of air in pipes is small, but the actual loss of power is still smaller. The pipe itself acts in a measure like a receiver—as a reservoir of power. It is probable that much of the transmission power loss experienced in practice is due to leakage from joints and flaws in the pipe.

Loss of Pressure or Head. For short distances the loss of pressure may be considered as taking place according to the laws governing the flow of all fluids, varying directly as the length of pipe, directly as the square of the velocity, and inversely as the diameter of the pipe. But for long distances the application of these laws becomes somewhat complex. In addition to the factors just given, it is necessary to take into account the volume and pressure of the air, and the difference between the pressures at the receiver and at the end of the pipe line. All are more or less interdependent. A statement of the case, more accurate than the above, is as follows: For a given diameter of pipe, when the volume of compressed air discharged and its initial pressure remain constant, the loss of pressure is proportionate to the length of the pipe.

But in actual service the initial pressure and the volume of discharge do not remain constant, and, in the passage of the air through the pipe, other modifying factors must be taken into account. In flowing through a long line of piping the pressure is gradually reduced by friction, while the volume is correspondingly increased. Therefore, to maintain in the pipe the flow of a given quantity of air whose volume is constantly increasing, the velocity also must increase, and this requires an increase of head or pressure.

The formulas commonly used are constructed on the hypothesis that the loss of head is proportional to the length of pipe, so that, if a certain head be required to maintain the flow of a given quantity of air in a pipe 1,000 feet long, twice this head would suffice for a pipe 2,000 feet long. But in this case, when the air has passed through the first thousand feet of pipe its motive head has been lost; and as the volume has thereby increased, a greater head will be necessary to maintain the flow in the second thousand feet. In other words, the ordinary formulas do not take into account the increase of volume due to the reduction of pressure, *i.e.*, loss of head.

To transmit a given volume of air at a uniform velocity and loss of pressure it would be necessary to construct the pipe with a gradually increasing area. This of course is impracticable, and if the rate of discharge is to be kept constant in pipe of uniform section, both volume and velocity must increase as the pressure is reduced by friction. The loss of head in properly proportioned pipes is so small, however, that in practice the increase in volume is usually neglected.

The actual discharge capacity of piping is not proportional to the cross-sectional area alone—that is, to the square of the diameter. Although the periphery is directly proportional to the diameter, the interior surface resistance is much greater in a small than in a large pipe, because as the pipe becomes smaller the ratio of perimeter to area increases. To pass a given volume of compressed air a 1-in. pipe of given length requires over 3 times as much head as a 2-in. pipe of the same length. The character of the pipe also, and the condition of its inner surface, have much to do with the friction developed by the flow of air. Besides imperfections in the surface of the metal, the irregularities incident upon coupling together the lengths of pipe must increase friction.

There are so few reliable data that the influences by which the values of some of the factors may be modified are not fully understood, and owing to these uncertain conditions the results obtained from formulas are only approximately correct. Among

the formulas in common use for determining the loss of pressure in pipes perhaps the most satisfactory is that of D'Arcy. As adapted for compressed-air transmission it takes the form:

$$D = c\sqrt{\frac{d^5(p_1 - p_2)}{w_1 l}}, \text{ or } D = \frac{c\sqrt{d^5}}{\sqrt{l}} \times \sqrt{\frac{p_1 - p_2}{w_1}}$$

in which

D = the volume of compressed air in cubic feet per minute discharged at the final pressure,

c = a coefficient varying with the diameter of the pipe, as determined by experiment,

d = diameter of pipe in inches,*

l = length of pipe in feet,

p_1 = initial gauge pressure in pounds per square inch,

p_2 = final gauge pressure in pounds per square inch,

w_1 = the density of the air, or its weight in pounds per cubic foot, at the initial pressure p_1.

The second form of the formula, as given above, will be found convenient for most calculations, as the factors can be considered in groups.

In the following table are given the values of c, d^5, and $c\sqrt{d^5}$. The values of c show some apparent discrepancy for sizes of pipe

TABLE XVII

Diameter of Pipe, Inches.	Values of c	Fifth Powers of d	Values of $c\sqrt{d^5}$
1	45.3	1	45.3
2	52.6	32	297
3	56.5	243	876
4	58.0	1024	1856
5	59.0	3125	3298
6	59.8	7776	5273
7	60.3	16807	7817
8	60.7	32768	10988
9	61.0	59049	14812
10	61.2	100000	19480
11	61.8	161051	24800
12	62.0	248832	30926

* The actual diameters of wrought-iron pipe are not the same as the nominal diameters for all sizes. This difference is small, however, except in the 1¼-in. and 1½-in. sizes, the actual diameters of which are 1.38 ins. and 1.61 ins. respectively.

larger than nine inches, but there would be no very material differences in the results.

Table XVIII gives the values of w_1 for initial gauge pressures up to 100 pounds per square inch:

TABLE XVIII

Gauge Pressure, Pounds.	w_1	$\sqrt{w_1}$	Gauge Pressure, Pounds.	w_1	$\sqrt{w_1}$
0	0.0761	0.276	55	0.3607	0.600
5	0.1020	0.319	60	0.3866	0.622
10	0.1278	0.358	65	0.4125	0.642
15	0.1537	0.392	70	0.4383	0.662
20	0.1796	0.424	75	0.4642	0.681
25	0.2055	0.453	80	0.4901	0.700
30	0.2313	0.481	85	0.5160	0.718
35	0.2572	0.507	90	0.5418	0.736
40	0.2831	0.532	95	0.5677	0.753
45	0.3090	0.556	100	0.5936	0.770
50	0.3348	0.578			

To facilitate computations in connection with D'Arcy's formula, Table XIX has been compiled by Mr. William Cox. It gives the values of $\sqrt{\frac{p_1-p_2}{w_1}}$ for terminal gauge pressures of from 20 to 100 lbs., and for pressure losses of from 1 to 10 lbs.*

Intermediate values can be obtained by interpolation. No allowance is made for pipe leakage, nor for incidental friction due to bends in the pipe.

By using these tables all ordinary problems involved in compressed-air transmission can be readily solved. For example, given a 5-in. pipe, 2,500 ft. long; how many cubic feet of air per minute at an initial pressure of 70 lbs. can be transmitted, with a loss of pressure of not more than 3 lbs.?

From Table XVII, $c\sqrt{d^5} = 3{,}298$; from Table XIX, $\sqrt{\frac{p_1-p_2}{w_1}} = 2.570$ and $\sqrt{l} = 50$. Substituting in the formula already given:

$$D = \frac{3{,}298}{50} \times 2.570 = 169.5 \text{ cu. ft. compressed air per minute.}$$

* Reproduced by permission from *Compressed Air*, Feb., 1898, pp. 374–376.

Table XIX

Values of $\sqrt{\dfrac{p_1 - p_2}{w_1}}$

Final Pressure p_2, lbs.	Losses of Pressure, $p_1 - p_2$.									
	1 lb.	2 lbs.	3 lbs.	4 lbs.	5 lbs.	6 lbs.	7 lbs.	8 lbs.	9 lbs.	10 lbs.
20	2.325	3.241	3.918	4.466	4.930	5.336	5.693	6.014	6.309	6.574
21	2.293	3.198	3.868	4.410	4.870	5.272	5.627	5.946	6.237	6.502
22	2.262	3.157	3.819	4.356	4.812	5.211	5.564	5.878	6.168	6.432
23	2.233	3.117	3.772	4.304	4.756	5.152	5.501	5.814	6.102	6.362
24	2.205	3.079	3.727	4.254	4.702	5.093	5.440	5.752	6.036	6.296
25	2.178	3.042	3.684	4.206	4.649	5.036	5.381	5.688	5.973	6.233
26	2.152	3.007	3.642	4.158	4.597	4.981	5.323	5.630	5.913	6.173
27	2.127	2.973	3.601	4.112	4.548	4.928	5.268	5.572	5.856	6.113
28	2.103	2.939	3.561	4.068	4.499	4.877	5.215	5.518	5.799	6.056
29	2.079	2.907	3.523	4.024	4.452	4.828	5.164	5.466	5.745	5.999
30	2.056	2.876	3.485	3.982	4.408	4.781	5.114	5.414	5.691	5.942
31	2.034	2.844	3.448	3.942	4.365	4.735	5.066	5.364	5.637	5.888
32	2.012	2.815	3.414	3.904	4.323	4.690	5.019	5.312	5.586	5.834
33	1.991	2.786	3.381	3.866	4.282	4.646	4.971	5.264	5.535	5.782
34	1.971	2.759	3.348	3.830	4.242	4.603	4.926	5.216	5.487	5.733
35	1.952	2.733	3.317	3.794	4.202	4.561	4.881	5.170	5.439	5.686
36	1.933	2.707	3.286	3.758	4.164	4.520	4.839	5.126	5.394	5.639
37	1.915	2.682	3.255	3.724	4.126	4.480	4.797	5.084	5.349	5.594
38	1.897	2.656	3.225	3.690	4.090	4.441	4.757	5.042	5.307	5.550
39	1.879	2.632	3.196	3.658	4.054	4.404	4.717	5.002	5.265	5.509
40	1.862	2.608	3.168	3.626	4.020	4.368	4.680	4.962	5.226	5.468
41	1.845	2.585	3.140	3.596	3.987	4.333	4.643	4.924	5.187	5.426
42	1.829	2.563	3.114	3.566	3.956	4.299	4.609	4.888	5.148	5.385
43	1.813	2.542	3.088	3.538	3.924	4.267	4.575	4.852	5.109	5.344
44	1.798	2.521	3.064	3.510	3.895	4.235	4.540	4.814	5.070	5.306
45	1.783	2.501	3.040	3.484	3.866	4.203	4.506	4.778	5.034	5.268
46	1.769	2.481	3.017	3.458	3.837	4.171	4.471	4.744	4.998	5.230
47	1.755	2.462	2.995	3.432	3.808	4.139	4.439	4.710	4.962	5.192
48	1.742	2.444	2.972	3.406	3.779	4.109	4.408	4.676	4.926	5.155
49	1.729	2.426	2.950	3.380	3.752	4.080	4.376	4.642	4.890	5.120
50	1.716	2.407	2.927	3.356	3.725	4.051	4.344	4.608	4.857	5.085
51	1.703	2.389	2.906	3.332	3.698	4.022	4.313	4.578	4.824	5.050
52	1.690	2.372	2.886	3.308	3.671	3.993	4.283	4.546	4.791	5.015
53	1.678	2.355	2.865	3.284	3.645	3.965	4.254	4.516	4.758	4.983
54	1.666	2.338	2.844	3.260	3.620	3.938	4.225	4.484	4.728	4.952
55	1.654	2.321	2.823	3.238	3.596	3.911	4.196	4.456	4.698	4.920
56	1.642	2.304	2.804	3.216	3.571	3.885	4.169	4.428	4.668	4.889
57	1.630	2.289	2.785	3.194	3.547	3.860	4.143	4.400	4.638	4.860
58	1.619	2.273	2.766	3.172	3.524	3.835	4.117	4.372	4.611	4.832
59	1.608	2.258	2.747	3.152	3.502	3.811	4.091	4.346	4.584	4.803
60	1.597	2.242	2.730	3.132	3.479	3.787	4.066	4.320	4.557	4.775
61	1.586	2.228	2.712	3.112	3.458	3.764	4.042	4.294	4.530	4.747
62	1.576	2.214	2.695	3.092	3.437	3.742	4.019	4.268	4.503	4.718
63	1.566	2.200	2.678	3.074	3.417	3.720	3.995	4.244	4.476	4.693
64	1.556	2.186	2.662	3.056	3.397	3.698	3.971	4.220	4.452	4.668
65	1.546	2.173	2.647	3.038	3.376	3.676	3.948	4.196	4.428	4.642
66	1.537	2.160	2.631	3.020	3.356	3.654	3.926	4.172	4.404	4.617

TABLE XIX—*Continued*

Values of $\sqrt{\dfrac{p-p_2}{w_1}}$

Final Pressure. p_2, lbs.	LOSSES OF PRESSURE, p_1-p_2.									
	1 lb.	2 lbs.	3 lbs.	4 lbs.	5 lbs.	6 lbs.	7 lbs.	8 lbs.	9 lbs.	10 lbs.
67	1.528	2.147	2.615	3.002	3.337	3.634	3.905	4.150	4.380	4.592
68	1.519	2.134	2.600	2.984	3.318	3.615	3.884	4.128	4.356	4.566
69	1.510	2.122	2.584	2.968	3.300	3.596	3.863	4.104	4.332	4.541
70	1.501	2.100	2.570	2.952	3.283	3.576	3.842	4.082	4.308	4.516
71	1.492	2.098	2.556	2.936	3.265	3.556	3.820	4.060	4.284	4.494
72	1.484	2.086	2.543	2.920	3.247	3.537	3.799	4.038	4.263	4.471
73	1.476	2.075	2.529	2.904	3.229	3.517	3.778	4.018	4.242	4.449
74	1.468	2.064	2.515	2.888	3.211	3.498	3.759	3.998	4.221	4.427
75	1.460	2.052	2.501	2.872	3.193	3.480	3.741	3.978	4.200	4.405
76	1.452	2.041	2.487	2.856	3.177	3.463	3.723	3.958	4.179	4.383
77	1.444	2.030	2.473	2.842	3.162	3.446	3.704	3.938	4.158	4.361
78	1.436	2.019	2.461	2.828	3.146	3.429	3.686	3.918	4.137	4.339
79	1.428	2.009	2.449	2.814	3.130	3.412	3.667	3.898	4.116	4.317
80	1.421	1.999	2.437	2.800	3.115	3.395	3.648	3.878	4.095	4.294
81	1.414	1.989	2.425	2.786	3.009	3.377	3.630	3.858	4.074	4.272
82	1.407	1.979	2.413	2.772	3.084	3.360	3.611	3.840	4.053	4.253
83	1.400	1.969	2.401	2.758	3.068	3.343	3.593	3.820	4.035	4.234
84	1.393	1.959	2.388	2.744	3.052	3.326	3.575	3.802	4.017	4.215
85	1.386	1.949	2.376	2.730	3.037	3.310	3.559	3.786	3.999	4.196
86	1.379	1.939	2.364	2.716	3.022	3.294	3.543	3.768	3.981	4.177
87	1.372	1.929	2.352	2.702	3.008	3.279	3.527	3.752	3.963	4.158
88	1.365	1.920	2.340	2.690	2.994	3.265	3.511	3.734	3.945	4.139
89	1.358	1.910	2.330	2.678	2.981	3.250	3.495	3.718	3.927	4.120
90	1.351	1.901	2.319	2.666	2.967	3.235	3.479	3.700	3.909	4.101
91	1.345	1.893	2.309	2.654	2.954	3.221	3.463	3.684	3.891	4.082
92	1.339	1.884	2.298	2.642	2.940	3.206	3.447	3.666	3.873	4.064
93	1.333	1.876	2.288	2.630	2.927	3.191	3.432	3.650	3.855	4.048
94	1.327	1.867	2.278	2.618	2.914	3.177	3.416	3.634	3.840	4.032
95	1.321	1.859	2.267	2.606	2.900	3.162	3.401	3.618	3.825	4.016
96	1.315	1.850	2.257	2.594	2.887	3.148	3.387	3.604	3.810	4.000
97	1.309	1.842	2.246	2.582	2.873	3.135	3.373	3.590	3.795	3.984
98	1.303	1.833	2.236	2.570	2.862	3.123	3.360	3.576	3.780	3.969
99	1.297	1.825	2.226	2.560	2.851	3.110	3.347	3.562	3.765	3.953
100	1.291	1.817	2.217	2.550	2.840	3.098	3.334	3.548	3.750	3.937

Volumes of compressed air are easily converted into corresponding volumes of free air by multiplying by the absolute pressure in terms of atmospheres (1 atmosphere = 14.7 lbs.). Thus, 100 cu. ft. of air at 80 lbs. gauge pressure, or 94.7 absolute pressure, are equal to 644 cu. ft. of free air, at sea-level. Table XIII gives the air pressures in pounds per square inch for

altitudes up to 15,000 ft., with the corresponding barometric readings.

Another formula for the loss of pressure in pipes has been published by Mr. Frank Richards, as follows:*

$$H = \frac{V^2 L}{10{,}000 D^5 a}$$

D = diameter of pipe in inches.

L = length of pipe in feet.

V = volume of compressed air delivered, in cubic feet per minute.

H = head or difference of pressure required to overcome friction and maintain the flow.

a = constant for diameter of pipe.

VALUES OF a FOR DIFFERENT NOMINAL DIAMETERS OF WROUGHT-IRON PIPE.

1"	0.350	3"	0.730	5"	0.934
1¼"	0.500†	3½"	0.787	6"	1.000
1½"	0.662†	4"	0.840	8"	1.125
2"	0.565			10"	1.200
2½"	0.650			12"	1.260

Using this formula with its constants, the calculated losses of pressure are smaller, and, conversely, the volumes of air discharged are larger, under the same conditions, than those obtained from D'Arcy's formula.

The losses of pressure in a table by F. A. Halsey indicate that the constants used by him differ materially from those given above. For comparison a series of random examples are shown in Table XX.

An examination of this table shows that in all cases the figures from D'Arcy's formula lie between the others, and until further experimental data are available it would appear safe to conclude that the results obtained from this formula are sufficiently ac-

* *American Machinist*, Dec. 27th, 1894.

† The values of a for 1¼- and 1½-in. pipe are not consistent with those for other sizes. See foot-note on page 251.

TABLE XX

Cubic Feet of Air at Final Pressure.	Length of Pipe, Feet.	Diameter of Pipe, Inches.	TRANSMISSION LOSSES, POUNDS.		
			Richards.	D'Arcy (Cox).	Halsey.
1,000	1,000	4	3.23	3.71	5.02
1,000	1,000	5	.95	1.17	1.63
1,000	1,000	6	.35	.46	.64
4,000	5,000	8	5.92	8.44	13.05
4,000	5,000	10	1.78	2.81	4.20
4,000	5,000	12	.68	1.06	1.70

curate for ordinary calculations. It must be remembered that, within certain limits, the loss of head or pressure increases with the square of the velocity. To obtain the best results it is found in practice that the velocity of flow in the main air pipes should not exceed twenty or twenty-five feet per second. Experiments made to determine the loss of pressure in the mains of the Paris compressed-air plant gave the following results: *

TABLE XXI
DIAMETER OF PIPE, TWELVE INCHES

Velocity of Flow in Feet per Second.	Initial Pressure, Pounds.	Final Pressure, Pounds.	Per Cent. of Initial Pressure Lost per Mile.
25	100	97.6	2.4
50	100	90.6	9.4
100	100	53.8	46.2

It is evident that when the initial velocity much exceeds 50 ft. per second the percentage loss becomes very large; and, furthermore, by using piping large enough to keep down the velocity the friction loss may be almost eliminated. For example, at the Hoosac tunnel, in transmitting 875 cu. ft. of free air per minute at an initial pressure of 60 lbs., through an 8-in. pipe 7,150 ft. long, the average loss including leakage was only 2 lbs. The velocity in this case was 8½ ft. per second. A volume of 500 cu. ft. of free air per minute, at 75 lbs. gauge pressure, can

* Unwin. Van Nostrand's Science Series, No. 106, p. 78.

be transmitted through 1,000 ft. of 3-in. pipe with a loss of 4.1 lbs., while if a 5-in. pipe were used the loss would be reduced to .24 lb., the velocities being respectively 28 ft. and 10 feet per second. In driving the Jeddo mining tunnel, at Ebervale, Luzerne Co., Penna., two 3½-in. machine drills were used in each heading, with a 6-in. main, the maximum distance of transmission being about 10,800 ft. This pipe was so large in proportion to the volume of air required for the 2 drills (about 230 cu. ft. free air per minute) that the loss was reduced to an extremely small quantity, the velocity being only 3⅓ ft. per second. A calculation shows a loss of .002 lb., and the gauges at each end of the main were found to record practically the same pressure.

A due regard for economy in installation, however, must limit the use of very large piping, the cost of which should be considered in relation to the cost of air compression in any given case. Diameters of from 4 to 6 ins. for the air mains are large enough for operating simultaneously from 6 to 10 drills. Up to a length of 3,000 ft. a 4-in. pipe will carry per minute 480 cu. ft. of free air compressed to 82 lbs., with a loss of 2 lbs. pressure. This volume of air will run four 3-in. drills. Under the same conditions a 6-in. pipe, 5,000 ft. long, will carry 1,100 cu. ft. of free air per minute, or enough for 10 drills in constant operation.

A mistake is often made in putting in branch pipes of too small a diameter. For a distance of, say, 100 ft. a 1½-in. pipe is small enough for a single drill, though 1-in. is frequently used. While it is, of course, admissible to increase the velocity of flow in short branches considerably beyond 20 ft. per second, extremes should be avoided. To run a 3-in. drill from a 1-in. pipe 100 ft. long would require a velocity of flow of about 55 ft. per second, causing a loss of 10 lbs. pressure. In this connection Table XXII* may be studied with advantage.

Compressed-Air Piping. The pipe for conveying compressed air may be of cast or wrought iron. If of wrought iron, as is customary, the lengths are connected either by sleeve couplings or by cast-iron flanges into which the ends of the pipe are ex-

* From the catalogue of the Norwalk Iron Works Co.

TABLE XXII

UNIFORM PRESSURE AT THE ENTRANCE OF THE PIPE, 80 LBS. GAUGE.

Nominal Size of Pipe. ☞	1 in.		1¼ in.		1½ in.		2 ins.		2½ ins.	
Length of Pipe in Feet. ☞	50	100	100	300	100	300	200	500	250	600
PRESSURES AT THE POINT OF DELIVERY.										
79.8	23.2	16.4	35.2	20.3	63.6	36.7	84.7	53.6	142.	91.7
79.6	33.1	23.4	49.7	28.7	89.9	51.9	119.6	75.7	200.9	129.6
79.4	40.4	28.6	61.0	35.2	109.1	63.0	146.5	92.7	244.4	157.7
79.2	46.8	33.1	70.3	40.6	127.1	73.4	169.1	107.1	283.2	183.1
79.	52.3	37.0	78.6	45.4	142.0	82.0	189.1	119.7	317.1	204.6
78.8	57.1	40.4	86.1	49.7	155.4	89.7	207.	131.0	348.4	224.8
78.6	61.6	43.6	93.0	53.7	168.0	97.0	223.3	141.3	377.0	243.9
78.4	65.9	46.6	99.2	57.3	179.3	103.5	238.7	151.1	399.6	258.4
78.2	70.3	49.7	105.4	60.8	190.5	110.0	252.9	160.1	424.1	273.6
78.	73.7	52.1	110.8	64.0	200.7	115.9	266.5	168.7	446.7	288.6
77.8	77.2	54.6	116.2	67.1	209.9	121.2	279.2	176.7	469.0	302.6
77.6	80.7	57.1	121.4	70.1	219.1	126.5	291.5	184.5	489.6	315.9
77.4	84.0	59.4	126.3	72.9	228.1	131.7	303.4	192.0	509.3	328.6
77.2	87.1	61.6	131.1	75.7	236.7	136.6	314.4	199.0	528.3	340.8
77.	90.3	63.7	135.4	78.2	245.2	141.6	325.5	206.0	546.5	352.6
76.8	92.9	65.7	139.8	80.7	252.4	145.7	336.1	212.7	564.2	364.0
76.6	95.6	67.7	143.9	83.1	259.8	150.0	346.2	219.1	581.3	375.0
76.4	98.4	69.6	148.1	85.5	267.6	154.5	356.0	225.3	597.5	385.5
76.2	101.0	71.5	152.1	87.8	274.7	158.7	365.6	231.4	613.8	396.0
76.	103.8	73.4	156.1	90.1	281.3	162.4	375.6	237.3	629.3	406.0
75.8	106.3	75.2	159.7	92.2	288.4	166.6	383.9	243.0	644.5	415.8
75.6	108.7	76.9	163.3	94.3	295.5	170.6	392.8	248.6	659.2	425.3
75.4	111.0	78.5	167.0	96.4	301.7	174.2	401.4	254.0	673.8	434.7
75.2	113.3	80.1	170.4	98.4	307.9	177.8	409.7	259.3	687.8	443.8
75.	115.5	81.7	173.9	100.4	314.3	181.5	417.9	264.5	701.6	452.7

CUBIC FEET OF FREE AIR DELIVERED AS COMPRESSED AIR AT THE STATED PRESSURES.

panded or screwed. Sleeve couplings are used for all except the large sizes. The smaller sizes, up to $1\frac{1}{4}$ in. are butt-welded, while all from $1\frac{1}{2}$ in. up are lap-welded to insure the necessary strength. Extra heavy piping may be had for higher pressures than those commonly used. Wrought-iron spiral-seam riveted, or spiral-weld steel, tubing is sometimes used. It is made in lengths of 20 ft. or less. For convenience of transport in remote regions rolled sheets in short lengths may be had. They are punched around the edges, ready for riveting, and are packed closely, 4, 6 or more sheets in a bundle.

All joints in air mains and branches should be carefully made. The pipe may be tested from time to time by allowing the air at full pressure to remain in the pipe long enough to observe the gauge. In case a leak is indicated it should be traced and stopped immediately. Air leaks are more expensive than steam leaks because of the losses already suffered in compressing the air. In putting together screw joints care should be taken that none of the white lead or other cementing material is forced into the pipe. This would cause obstruction and increase the friction loss. Also, each length as put in place should be cleaned thoroughly of all foreign substances which may have lodged inside. To render the piping readily accessible for inspection and stoppage of leaks it should, if buried, be carried in boxes sunk just below the surface of the ground; or, if underground, it should be supported upon brackets along the side of the mine workings. Low points in pipe lines, which would form "pockets" for the accumulation of entrained water, should be avoided, as they obstruct the passage of the air. In long pipe lines, where a uniform grade is impracticable, provision may be made near the end for blowing out the water at intervals, when the air is to be used for pumps, hoists, or other stationary engines.

For long lengths of piping expansion joints are required, particularly when on the surface. Underground they are not often necessary, as the temperature is usually nearly constant, except in shafts, or elsewhere, where there may be considerable variations of temperature between summer and winter.

As each bend or elbow in a pipe line has a serious retarding effect, abrupt changes in direction and sharp curves should be avoided so far as possible. For the same diameter of pipe the resistance caused by a bend increases as the radius of the curve diminishes, but the exact relation is not accurately known. In the absence of sufficient experimental data the following table is given, as published in the catalogue of the Norwalk Iron Works Co.:

TABLE XXIII

Radius of elbow in terms of diameter of pipe............	5	3	2	1½	1¼	1	¾	½
Equivalent length of straight pipe in terms of its diameter..	7.85	8.24	9.03	10.36	12.72	17.51	35.09	121.2

It would appear that these allowances are none too large, since for steam piping the frictional resistance of each ordinary sharp right-angled elbow is considered equivalent to that due to a length of straight pipe equal to forty times its diameter. However, in putting in wrought-iron air piping of the sizes customarily used the bends are not necessarily so sharp as a standard right-angled elbow. When many sharp bends are permitted, it is evident that the resistance may become very great.

Under most conditions this difficulty may be avoided by the exercise of proper care in the installation of the pipe lines. The matter should have special consideration in the stopes of mines timbered with square sets. As far as possible, the piping should be carried diagonally through the sets, bending the pipe itself whenever necessary, instead of using right-angled elbows.

CHAPTER XVII

COMPRESSED AIR ENGINES

COMPRESSED air may be employed as a motive power in an engine in two ways, *viz:* at full pressure or expansively. By working at full pressure it is understood that the air is admitted to the cylinder throughout practically the entire length of stroke, *i.e.*, without cut-off, and that therefore nearly a cylinderful of air at gauge pressure is exhausted at each stroke. In this case the work of the air engine is roughly similar to that done in a non-expansive-working steam engine. Among the machines which use air in this way are rock-drills and simple, direct-acting pumps, without rotary parts.

By the term expansive-working it is meant that the air is admitted to the cylinder during only a part of the stroke, and is then cut off and thestroke completed by the expansive force of the air. For operating in this way some equalizing agent, such as the fly-wheel, is essential, and as a rule a higher initial pressure is employed than when working under full pressure throughout the stroke. It is necessary to distinguish between complete and partial or incomplete expansion. When the air is used with complete expansion the operation in the cylinder is the reverse of adiabatic compression in a compressor, the final pressure being equal to that of the atmosphere. But as no condensation is possible with air, it follows that the lowest terminal pressure in the cylinder must still be sufficiently above atmospheric pressure to produce a proper exhaust, and to overcome the friction of the engine at the end of the stroke. Hence, theoretically complete expansion is impracticable for simple air engines of ordinary design.

Most air engines work with partial or incomplete expansion,

the air expanding adiabatically in the latter part of the stroke. The point of cut-off is such that the terminal cylinder pressure exceeds the back-pressure by an amount sufficient to cause a free exhaust. In the conditions here set forth, no reference is made to the thermal changes incident upon adiabatic expansion in the air cylinder. Although in principle compressed air is used like steam, both being elastic fluids, there is an essential difference in the results obtained, due to the reduction in temperature. In expanding behind the piston, a given volume of compressed air at a given pressure will not produce the same amount of power as steam under the same conditions. If two curves be constructed, representing the expansion of equal volumes of air and steam, from the same initial pressure down to pressures below that of the atmosphere, it will be seen that the steam pressure at all points of the stroke is considerably higher than the air pressure; and the expansion curve of the air reaches the atmospheric line much sooner than the steam curve.

Fig. 122 shows an ideal card, in which the initial pressure is 75 lbs., and the cut-off is at $\frac{1}{6}$ stroke. The adiabatic expansion curve of the air shows that the pressure is reduced to zero gauge pressure when the air has expanded to $3\frac{3}{4}$ times the initial volume, the mean effective pressure being 18.9 lbs. At the end of the stroke the pressure falls to 7 lbs. below atmospheric pressure. The steam curve, on the other hand, does not cut the atmospheric line until the expansion reaches $4\frac{1}{3}$ times the initial volume, and the mean effective pressure is 25.2 lbs. The lower mean pressure of the air is due to the development of cold during its expansion. The operation is the reverse of compression, and the resulting loss of motive power is analogous to the loss of work in the compressor caused by the generation of heat. Just as the heat of compression reacts upon the air while being compressed in the cylinder, and produces a higher tension than that due to the mere reduction in volume; so conversely, when expansion takes place, the air, which is usually at normal atmospheric temperature on entering the cylinder, rapidly gives up its sensible heat, and the cold reacting upon the expanding air reduces its pressure faster

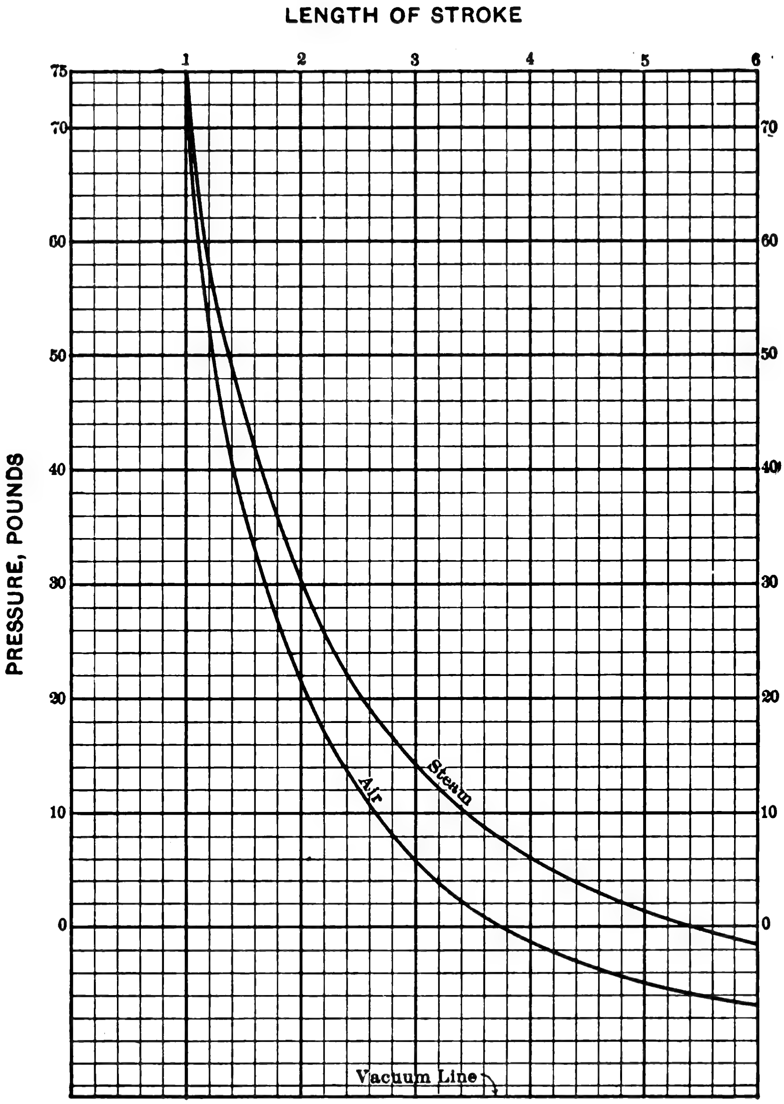


Fig. 122.—Expansion Curves of Steam and Air.

than that which is due to the increase in volume alone. Moreover, this behavior of compressed air is independent of the initial temperature, since the resulting expansion curve would be unaltered. In the case of steam the initial temperature is high, and is reduced but little during expansion from ordinary working pressures down to atmospheric pressure.

A similar comparison may be made for other initial pressures and ratios of cut-off. In every case the mean effective pressure is higher for steam than for air. It follows that, to develop the same amount of power in a given cylinder and with the same initial pressure, the cut-off must be later in the stroke with air than with steam.

So low are the temperatures produced by the expansion of air, from ordinary working pressures of sixty or seventy pounds down to atmospheric pressure, that for a long time the expansive use of compressed air was considered impracticable. In Table XXIV are given the theoretical final temperatures of the exhaust air, in working with complete expansion, and also at full pressure throughout the stroke, for different ratios of initial to final pressure, together with the theoretical efficiencies. The initial temperature is taken as 68° F.*

TABLE XXIV

Ratio of Initial to Final Pressure.	Working With Complete Expansion.		Working at Full Pressure.	
	Final Temperature. Degrees Fah.	Theoretical Efficiency.	Final Temperature. Degrees Fah.	Theoretical Efficiency.
2	— 28.2	.855	— 8.4	.82
3	— 76.	.806	—34.5	.72
4	—106.6	.782	—45.7	.67
5	—128.2	.768	—54.4	.63
6	—144.4	.758	—59.8	.60
7	—158.8	.751	—63.4	.57
8	—170.8	.746	—66.1	.55
9	—180.6	.742	—68.	.53
10	—189.2	.739	—69.7	.51

* M. Mallard, "Etude Théorétique sur les Machines à Air Comprimé," p. 27. Robert Zahner, "Transmission of Power by Compressed Air," p. 100.

In the table it is shown that by working at full pressure extremely low temperatures of exhaust are avoided; but the efficiency of this method of using compressed air is necessarily much below that obtained from expansive working. It is understood that the temperatures here given are theoretical and are never actually reached in practice. The cold produced is modified by several causes: (1) Some heat is transmitted from the external atmosphere through the cylinder walls; (2) the re-compression of the clearance air at each stroke produces heat in the cylinder, to a degree that increases with the initial pressure and the clearance volume; and (3) the presence of even a small quantity of moisture in the air tends in some degree to raise the cylinder temperature.

A few brief notes will here be given concerning the elements of the operation of compressed-air engines, that may be considered more or less applicable for ordinary service, *viz:* working at full pressure, with partial expansion, or with complete expansion. Isothermal expansion may be neglected, since it involves the application of a sufficient degree of external heat to the air while doing its work in the cylinder to produce a terminal temperature equal to the initial temperature.

1. **Working at Full Pressure.** This mode of using compressed air is common for engines like pumps, operating under a constant resistance and not provided with fly-wheels:

Let P' = the absolute initial pressure of the air.

V' = the initial volume of air, at the pressure P', or K times the volume of one pound of air used per unit of time.

T' = the absolute initial temperature of the compressed air.

T = the absolute final temperature of the air at exhaust, on expanding to atmospheric pressure.

P = pressure of the air at exhaust.

W = foot-pounds of work done.

From the theory of compressed air:

$R = J\,(C_p - C_v) = 778\,(0.2375 - 0.1689) = 53.37$, where J is Joule's heat unit, and C_p and C_v are the specific heats of air at constant pressure and constant volume.

As no work is done by the expansive force of the air originally produced by compression, W equals the volume of air used, V', multiplied by the difference between P' and P, or: $W = V'(P' - P)$.

Substituting for V' its value, $\frac{KRT'}{P'}$, as obtained from: $P' V' = KRT'$,

$$W = \frac{KRT'}{P'}(P' - P) = KRT'\left(1 - \frac{P}{P'}\right)$$

Giving R its value, 53.37: $W = 53.37\ KT'\left(1 - \frac{P}{P'}\right)$

2. **Working with Partial Expansion.** The advantages of using compressed air in this way may be obtained from engines possessing fly-wheels, provided that the cut-off be not too early in the stroke to avoid excessive reduction of cylinder temperature, or else that the air be reheated before entering the cylinder.

In this case the values of P', V', and T' are as above. From the point of cut-off the air expands adiabatically down to a terminal pressure of P'' and volume V'', the final temperature in the cylinder falling to T''. On exhausting, the pressure, volume, and temperature become P, V, and T. The work done is composed of three parts, *viz:*

W' = work between the point of admission and the point of cut-off $= P'\ V'$.

W'' = work performed by expansion of the volume V' from the point of cut-off to the end of the stroke $= 778\ KC_v\ (T' - T'')$.

W''' = negative work due to back-pressure $= -P\ V''$.

Taking the algebraic sum of these three quantities:

$$W = P'\ V' + 778\ K\ C_v\ (T' - T'') - P\ V''$$

But, as under (1): $V' = \frac{KRT'}{P'}$ and $V'' = \frac{KRT''}{P''}$

Substituting these values of V' and V'', and for R and C_v their numerical values of 53.37 and 0.1689:

$$W = K\left[53.37\ T' + 131.4\ (T' - T'') - 53.37\ T'\left(\frac{P}{P''}\right)\right]$$

$$= 53.37\ K\left[T' + 2.46\ (T' - T'') - T'\frac{P}{P''}\right]$$

3. **Working with Complete Expansion.** In the theoretical card, Fig. 123, are shown the relations of the compression and expansion lines, the shaded portion representing the useful work done by the complete expansion of cold air in a motor cylinder.

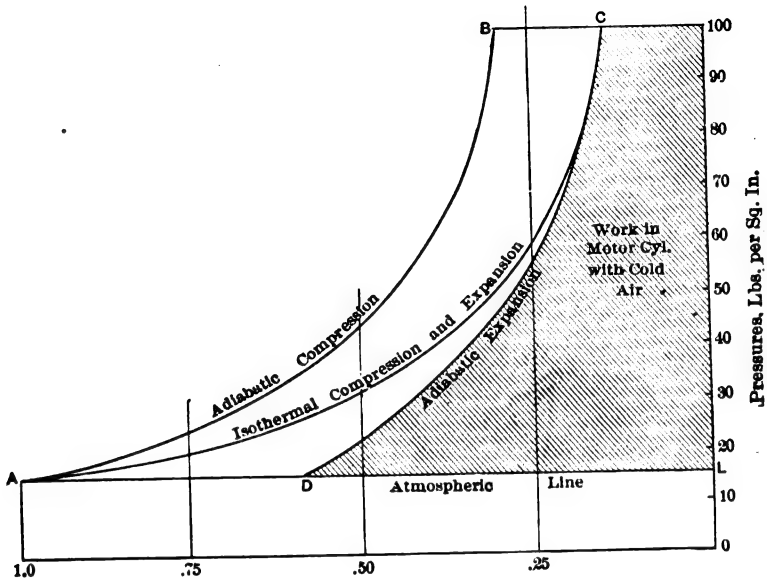


FIG. 123.

When the expansion is adiabatic, the same relations exist between pressures, volumes, and temperatures as were set forth in the discussion of adiabatic compression, *viz:*

$$\frac{P'}{P} = \left(\frac{V}{V'}\right)^{n=1.406} = \left(\frac{T'}{T}\right)^{\frac{n-1}{n}=0.29}$$

The theoretical work done by complete adiabatic expansion may be expressed by a formula similar to that employed for compression, but with an inversion of certain of the quantities, thus:

$$W = \frac{n}{n-1} PV \left[1 - \left(\frac{P}{P'}\right)^{\frac{n-1}{n}}\right], \text{ in which}$$

W = the theoretical foot-pounds of work done by the expan-

sion to atmospheric pressure of 1 pound (13.1 cu. ft.) of free air. Substituting the values of the constants, and for working at sea-level:

$$W = 3.463 \times 144 \times 14.7 \times 13.1 \times$$

$$\left[1-\left(\frac{14.7}{P'}\right)^{0.29}\right] = 96{,}029\left[1-\left(\frac{14.7}{P'}\right)^{0.29}\right]$$

For example, if P' be 40 lbs. gauge pressure:

$$W = 96{,}029\left[1-\left(\frac{14.7}{54.7}\right)^{0.29}\right] = 30{,}440 \text{ ft. lbs., or } 2{,}323 \text{ ft. lbs.}$$

per cu. ft. of free air.

Actual Work Done. In the above expressions no account is taken of the friction of moving parts of the motor engine, or loss of work caused by leakage. In determining the actual work, the general case will be where a cut-off is employed. The relations between initial and terminal pressures and temperatures, for different ratios of expansion in a motor-engine cylinder, are shown in Table XXV,* the points of cut-off, in tenths of the cylinder stroke, being given in the first column.

The quantities in Table XXV must be further corrected for piston clearance and the lost volume represented by the air ports and passages of the cylinder, because part of the air expands into these clearance spaces. Therefore, the actual effect of the cut-off, in any given case, is found by dividing the sum of the cut-off plus clearance, by the cylinder volume plus clearance. For example, if the stroke be 10, with a cut-off of $\frac{4}{10}$, and clearance of 6 per cent., the actual volume of the cylinder, including clearance, will be: $(10 \times .06) + 10 = 10.6$. Then the ratio of actual cut-off, plus clearance, is $4 + .6 = 4.6$, and the working cut-off becomes $4.6 \div 10.6 = 0.434$. In this way Table XXVI has been constructed, for use in connection with Table XXV. It shows the actual cut-off corresponding to the different nominal points of cut-off, for the percentages of piston clearance named at the top of the columns.

* This table, as well as Table XXVI, is taken in part from those used by Gardner D. Hiscox, in "Compressed Air, its Production, Uses and Application," 1901, p. 202.

TABLE XXV

THEORETICAL RATIOS OF PRESSURES AND TEMPERATURES DUE TO THE EXPANSION OF COMPRESSED AIR IN A MOTOR CYLINDER.

Cut-off.	Ratio of Expansion = 1 ÷ Cut-off.	Ratio of Mean to Total Absolute Pressure, for Entire Stroke.	Ratio of Mean to Total Absolute Pressure, During Expansion only.	Ratio of Initial to Final Temperature.	Ratio of Initial to Final Absolute Temperature, Due to Expansion only.	Ratio of Initial to Final Absolute Pressure for Ratio of Expansion.
0.10	10.00	0.249	0.166	0.391	0.513	0.039
.15	6.67	.348	.233	.460	.578	.069
.20	5.00	.436	.295	.518	.627	.104
.25	4.00	.515	.353	.568	.669	.142
.30	3.33	.585	.408	.612	.705	.184
.35	2.86	.647	.460	.652	.737	.228
.40	2.50	.706	.510	.688	.767	.275
.45	2.22	.757	.558	.722	.794	.325
.50	2.00	.802	.604	.754	.818	.378
.55	1.81	.842	.649	.784	.841	.433
.60	1.67	.877	.692	.812	.862	.487
.65	1.54	.907	.734	.839	.882	.545
.70	1.43	.932	.774	.865	.902	.605
.75	1.33	.954	.814	.889	.920	.667

TABLE XXVI

EXCESS OF CUT-OFF DUE TO PERCENTAGE OF CLEARANCE FOR THE NOMINAL CUT-OFFS IN COLUMN 1.

Nominal Cut-off.	Percentage of Clearance.						
	.03	.04	.05	.06	.07	.08	.10
.10	0.126	0.135	0.143	0.151	0.159	0.167	0.182
.15	.175	.184	.191	.198	.206	.213	.227
.20	.223	.231	.238	.245	.252	.259	.273
.25	.272	.279	.286	.293	.299	.305	.318
.30	.320	.327	.333	.340	.346	.352	.364
.35	.368	.376	.380	.387	.392	.398	.409
.40	.417	.423	.429	.434	.439	.444	.455
.45	.465	.471	.477	.481	.486	.490	.500
.50	.514	.519	.524	.528	.533	.537	.546
.55	.564	.568	.571	.576	.580	.585	.591
.60	.612	.615	.619	.623	.626	.630	.637
.65	.660	.664	.667	.670	.673	.676	.682
.70	.709	.711	.714	.717	.720	.722	.727
.75	.758	.760	.762	.764	.766	.768	.772

The theoretical terminal cylinder pressure resulting from adiabatic expansion may be expressed by:

$\frac{P'}{C^{1.406}} - P$, in which C = ratio of expansion = $\frac{1}{\text{point of cut-off}}$ (see column 2, Table XXV).

For example, for a cut-off at $\frac{4}{10}$ stroke and 65 lbs. gauge pressure, the terminal pressure (above atmospheric pressure) will be:

$$\frac{65 + 14.7}{2.5^{1.406}} - 14.7 = 7.2 \text{ lbs.}$$

The volume corresponding to the nominal cut-off is increased by the clearance, and adds to the mean pressure. Thus, in the above example, assuming the clearance to be 6 per cent., the actual cut-off (Table XXVI) is increased from 0.4 to 0.434, of which the ratio, C, is $\frac{1}{.434} = 2.3$. From Table XXV, column 7, the ratio of initial to terminal pressure, corresponding to the actual cut-off of 0.434, is (by interpolation) .31; whence: $(79.7 \times 0.31) - 14.7 =$ 10 lbs. terminal pressure.

Cylinder Volume Required for a Given Power. The work per stroke is found by dividing the foot-pounds of work to be done per minute by twice the number of revolutions of the engine (which would be determined for any given size of engine by the ordinary empiric rules of practice). This is substituted, with the initial and final pressures, in the formula for working with full pressure, partial or complete expansion, as the case may be, which is then solved for the initial volume, V′, of compressed air used per stroke. To the theoretical cylinder volume thus found, the allowance for piston clearance is added, according to the type of engine. The proper proportion between stroke and diameter of cylinder is finally determined.

The volume of free air per minute, required for an air engine, per indicated horse-power and for different ratios of cut-off, are shown in Table XXVII, by F. C. Weber.* The figures given in

* *Compressed Air*, Oct., 1896, p. 117.

this table do not include the volume corresponding to piston clearance which may be found as already shown.

Table XXVII

Cubic Feet of Free Air per Minute Used in Motor Engine, Per I. H.-P.

Point of Cut-off.	Gauge Pressures, Pounds.									
	30	40	50	60	70	80	90	100	110	125
1	23.3	21.3	20.2	19.4	18.8	18.42	18.10	17.8	17.62	17.40
¾	18.7	17.1	16.1	15.47	15.0	14.6	14.35	14.15	13.98	13.78
⅔	17.85	16.2	15.2	14.5	14.2	13.75	13.47	13.28	13.08	12.90
½	16.4	14.5	13.5	12.8	12.3	11.93	11.7	11.48	11.30	11.10
⅓	17.5	15.2	12.9	11.85	11.26	10.8	10.5	10.21	10.02	9.78
¼	20.6	15.6	13.4	13.3	11.40	10.72	10.31	10.0	9.75	9.42

In this table the air is supposed to be used without reheating, and at an initial temperature of 60° F. Reheating will reduce the volume of air proportionally to the ratio $\frac{T_2}{T_3}$, where $T_2 = 459° + 60° = 519°$ F., or absolute temperature; and $T_3 = 459°$ plus the temperature of the reheated air on entering the motor cylinder. Thus, if the air be reheated to 200° F., the above ratio becomes $\frac{519°}{659°} = 0.787$, by which decimal the volume of air as found in the table must be multiplied.

So far as mine service is concerned, it has been customary to consider compressed air almost exclusively as an agent for the operation of rock-drills, and in view of its preponderating application to this use its adaptability under proper conditions to the driving of other machines and engines is sometimes overlooked. Of late years, however, with improved methods of compression and reheating, attention has been given to employing compressed air for a greater variety of service; not only underground, but for certain portions of the surface plant of mines as well. Aside from cases where the disposal of exhaust steam would be

troublesome, the question is largely one of comparative loss in transmission and the power cost of the air.

Although not strictly in place in this chapter, reference may be made to what has been called the "two-pipe system" or "high-range compressed-air transmission," introduced some years ago by Charles Cummings.*

The machine or engine using the air makes in effect a closed circuit with the compressor. After the air has done its work in the motor cylinder, it is returned to the compressor at the pressure of the exhaust, through a second line of piping. The return pipe connects with a closed chamber at the compressor, in which the inlet valves are placed, thus enabling the compressor to begin its stroke with the cylinder filled under a considerable initial pressure. Then, after raising the pressure to the original point, the compressor delivers the air into the main, to be used again by the air engine. The actual working pressure of the air engine is, therefore, the difference between the pressures in the delivery and return pipes. Barring leakage, the same air is thus used over and over, the intention being that the compressor shall put back into the air kept in circulation the power expended in the motor engine cylinder.

Though the compressor itself is not materially different from the ordinary forms, the two-pipe system requires a rather complicated arrangement of piping and valves for charging the apparatus with air at the working pressure adopted, and for governing the speed and output according to the rate of consumption of air.† The advantages of the system are: a higher efficiency than is obtained from moderate-size compressors of the usual types, and less trouble from freezing at the motor engine by reason of the relative dryness of the air due to its higher tension. The efficiency increases with the pressure employed. In using compressed air without reheating the two-pipe system

* Patent No. 456,941 was issued to Mr. Cummings in 1891.

† A detailed illustrated description is given by Frank Richards in *American Machinist*, April 28th, 1898, p. 23. See also *Compressed Air Magazine*, Oct., 1907, p. 4599.

is superior in principle to the ordinary mode of operating compressed-air plant. But because of the greater first cost its advantages disappear when reheating can be adopted, and the single-pipe system is then found to be preferable.

The two-pipe system is best suited for machines working at full pressure throughout the stroke, such as machine drills or simple, direct-acting pumps. When the motor works expansively the pulsations become objectionable, as a regular flow of air is not maintained in the return pipe. Under these conditions the inertia and friction of high-pressure air in long pipe lines becomes noticeable and disadvantageous.

As the length of air pipe required for this system is doubled, not only may the first cost of the pipe go far toward offsetting the greater efficiency but, with at least twice as many joints in the pipe lines, the chances of loss from leakage are increased. And if very high pressures be used (pressures of several hundred pounds have been proposed), not only must the piping itself be heavier and more expensive, but the proportionate power loss from leakage is greater. For moderate distances, however, and when working at full pressure under the proper conditions, the foregoing disadvantages may be more than counterbalanced by the superior efficiency of the system. Though not yet in general use, the two-pipe system is said to have given satisfaction at several mines in New Mexico, Colorado, and California,* and has recently (1905) been proposed for use in the Johannesburg gold district. Some prominence is here given to the system because of its novel features and the probability that it may be found useful, if its disadvantages can be overcome. Reference may be made to a paper by H. C. Behr, published in 1905 in the *Transactions of the Mechanical Engineers' Association of the Witwatersrand*, in which the Cummings system is treated at length, with a discussion of its advantages as applied to compressed-air-driven pumps.

* A. E. Chodzko, *Modern Machinery* (Chicago), Jan., 1899, p. 11.

CHAPTER XVIII

FREEZING OF MOISTURE DEPOSITED FROM COMPRESSED AIR

Reference has been made in a former chapter to the trouble sometimes caused by the congelation of the moisture carried in compressed air when deposited in the transmission pipes or in the ports and exhaust passages of the machine using the air. The presence of moisture in compressed air must be accepted as an unavoidable condition. Existing in the atmosphere at all times in greater or less quantity, when air is compressed the moisture is carried with it. A part of the water is deposited in the air receiver, but a considerable quantity still remains and will be brought into evidence when the proper conditions occur.

The capacity of air for moisture depends primarily upon its temperature. Under ordinary atmospheric conditions 1,000 cubic feet of air contain about one pound of water. When its volume is reduced in the compressor cylinder, the increase of heat which takes place augments its moisture-carrying capacity. Any subsequent decrease in temperature reduces this capacity, and if the air be saturated the excess of moisture is deposited. Volume for volume, the capacity of air for moisture is independent of its pressure or density. That is, at the same temperature, a cubic foot of air at atmospheric pressure will hold in suspension the same weight of water as a cubic foot at 100 pounds pressure. But this must not be misunderstood. If a certain volume of moist atmospheric air be compressed isothermally, say to $\frac{1}{10}$ of its original volume, its water capacity is also reduced to $\frac{1}{10}$, and $\frac{9}{10}$ of the water originally present in the air is deposited. Therefore, while the capacity for carrying moisture of a given volume of air varies with the temperature, it must change also with any increase or decrease of pressure which changes its volume.

Causes of Freezing. Certain conditions are required to cause freezing of compressed air: deposited moisture must be present, and it must be subjected to a temperature below the freezing-point. So long as the temperature does not fall low enough, the presence of moisture can do no harm. Although one of the recognized functions of the air receiver is to permit the deposition of water before the air passes into the pipes, still, unless the receiver be extremely large, the air leaves it warm—usually even quite hot—and therefore carries with it considerable moisture. In the case of wet compressors, unless liberal sprays are used to attain effective cooling, the air is apt to contain more moisture than that from dry compressors. A well-designed injection compressor, however, not too small for its work and therefore running at a moderate speed, will deliver cool air which will not give trouble from freezing. The air having attained nearly normal temperature before entering the pipe-line, its moisture-carrying capacity undergoes but little further reduction while passing through the pipe, and only a small amount of additional deposition takes place. With dry compression the percentage of humidity of the intake air, and the temperature at discharge, determine the quantity of water carried out of the cylinder. The humidity, in turn, varies with the weather. Changes in the weather may quickly be followed by variations in the quantity of moisture deposited in the receiver and pipe-line. When the air is finally expanded in doing its work in the air engine, intense cold is produced as the pressure falls, and the latent heat of compression is absorbed. It is here that the moisture carried with the air into the pipes makes its appearance as frost and causes trouble. Watery vapor itself, depositing a light, snow-like frost, does not tend to clog the air passages and ports as much as entrained water in a finely divided state, which will gradually form accumulations of solid ice and choke the exhaust wholly or in part.

Prevention of Freezing. The difficulties which may arise from the conditions just outlined are apt to be exaggerated. That freezing not infrequently occurs is true, but with a properly

designed and arranged plant it may easily be avoided. Two things require attention: *first*, the air should be caused to drop its moisture as completely as possible before entering the main; *second*, provision should be made for draining off what deposited moisture remains in the pipe-line, before the air passes to the machine in which it is to be used. Although this is a simple matter, the means for accomplishing it are often neglected. Considerable quantities of water may collect in low places in the pipe-line and, if not blown out at intervals, will be carried into the ports, cylinder, and exhaust passage of the air machine and there freeze.

Granting that the air leaves the receiver near the compressor practically saturated and still warm, it is evident that a great improvement in working conditions may be realized by introducing a second receiver as close as possible to the machines using the air. In mining the second receiver is, of course, placed underground.* Before reaching it, the temperature of the air will have become normal, and the entrained moisture from the pipe-line may readily be trapped and drawn off. It may be remarked that automatic water-traps are preferable to valves or cocks for getting rid of the water. As a rule, when the compressed air is to be used expansively, a special aftercooler should be introduced, placed as close as possible to the compressor. In any case, the receiver should be of ample size to insure the deposition of the moisture. The advantages of reheating the air before use will be taken up later.

Influence upon Freezing of High Pressures in Transmission. The statements made in the first part of this chapter suggest an important consideration, *viz:* in transmitting power by air at a high pressure there is less liability to trouble from freezing than when low pressures are employed, provided that the length of pipe-line is sufficient to allow the air to be completely cooled and drained of its water while still under high pressure. At a low pressure a greater volume of air is required to furnish a given amount of power than when at a high pressure. More moisture

* See Chapter XI.

must, therefore, be dealt with, and at the low pressure it cannot be so thoroughly separated before the air is used. Suppose the transmission to be at a high pressure, and through a pipe long enough to allow the air to reach normal temperature. If the deposited moisture be drained away while the air is at its maximum pressure; then, if the air be subsequently expanded down to a lower pressure suitable for working (with a corresponding increase of volume) and allowed to regain its normal temperature, the percentage of moisture will be reduced, so that the air may be relatively very dry. When finally used in the air engine there will not be enough moisture present to cause troublesome freezing.

Deposition of Moisture by Reducing Pressure. Still another mode of minimizing trouble from freezing is to reduce the pressure of the air before it enters the cylinder of the air engine. The means by which this is accomplished and the results obtained may be illustrated by an example.

At the Drummond Colliery, Nova Scotia, for running an underground pump by compressed air two receivers are used, one near the pump, and another 300 ft. farther back on the pipeline. The air pressure in the main from the surface is 85 lbs., and as the proportions of the cylinders of this particular pump are such that so high a pressure was unnecessary a reducing valve was put in the pipe just before reaching the first receiver. By this valve the air is wire-drawn to reduce the pressure to forty-five pounds, which results in a deposition of nearly one-half the entrained water, in addition to that already deposited in the pipes. It is found that more moisture collects in the first than in the second receiver, and by this device the serious difficulty previously encountered from freezing at the pump has been entirely overcome.* The temperature lost by the reduction of pressure to forty-five pounds is regained before the air reaches the pump.

* This information has been kindly furnished by Charles Fergie, superintendent of the Drummond Colliery. See also Mr. Fergie's article on the subject, in *Transactions Canadian Mining Inst.*, 1896, of which an abstract was published in the *Colliery Guardian*, October 30th, 1896, p. 821.

Protection of Surface Piping. What precedes refers only to the freezing produced by internal reduction of temperature, acting on the moisture carried in the air. In using compressed air, even for mining purposes, it often becomes necessary to carry lines of air pipe considerable distances on the surface. To prevent condensation and freezing of the moisture in winter by external cold, all surface piping must be protected. If exposed to temperatures below the freezing-point, the inside of the pipe will become coated with ice and its effective cross-section reduced. A serious diminution of area may thus be caused at low points in the pipe-line, where water tends to collect; or the pipe may even be frozen solid in such places by the gradual accumulation of ice. Underground the temperature is rarely, if ever, low enough to render any protection necessary, except in cold, down-cast shafts, or in tunnels in which there is a strong inward draught.

Some time ago, at one of the Butte copper mines, a simple and inexpensive device was employed to prevent the freezing of moisture in a long line of surface piping. The air main of a large compressor plant was carried on the surface some hundreds of feet before reaching the shaft. During the winter months it was at times difficult to get sufficient air pressure in the mine because of the partial choking up of the pipe. As the volume of compressed air was too large to be dealt with by the ordinary receiver, a series of old tubular boilers were placed close to the compressor house. The hot air, at eighty pounds gauge pressure, in passing through these boilers, from one to another, was cooled down practically to atmospheric temperature and as a consequence a large part of its moisture was deposited. It was found that discarded tubular boilers, when strong enough, were well suited to this purpose, because of the large surface presented to the cold outside air; especially when they are set horizontally, so that there is a free circulation of air through the tubes. A blower might be used for the same purpose in a warm climate, or the boilers submerged in cold water. This effectual remedy is worthy of adoption where the conditions are similar.

CHAPTER XIX

REHEATING COMPRESSED AIR

After the warm compressed air leaves the compressor and receiver on its journey through the transmission line its temperature is quickly reduced to that of the surrounding atmosphere. The loss thus suffered could be prevented only by using the air immediately and before it has time to cool. But this is never possible in mining practice. It would be unreasonable to produce compressed air for use close to the compressor, because of the loss that inevitably ensues whenever power is converted from one form into another. The principal object in compressing air is to convert the power into a convenient form for transmission to a distance. The facility with which the heat of compression is given up, however, suggests that a gain may be effected by reheating the compressed air when it reaches the place where it is to be utilized.

The process is a simple one, and by such reheating an additional volume of air is obtained at a much lower power cost than if it were produced in the compressor itself. This may be shown by comparing the number of heat units required to produce a given volume of air at a given pressure in a compressor cylinder, with the heat units required to accomplish the same result by causing the air to expand through the direct application of heat. Herein is the ultimate basis of comparison for determining the useful effect of reheating.

Appliances for, and Results of Reheating. A number of methods of reheating have been actually used or proposed, the most important of which are as follows: (1) The air to be heated is passed through a cast-iron chamber or coil of pipe, exposed to a fire or current of hot gases or steam; (2) heat may be added

within the body of air itself, such as by the combustion of fuel, the injection of steam or hot water, or the placing in the air pipe of an electric-resistance coil.

The method most frequently employed is the one first named; it is preferable from a mechanical standpoint and is the most efficient. Those appliances in which internal combustion is adopted, or in which hot water or steam is the heating agent, are less satisfactory in practical operation, but are useful where the burning of fuel is not admissible.

The following calculation,* showing the results theoretically obtainable by reheating, presents the matter in concise form:

Weight of 1 cu. ft. of steam, at 75 lbs. gauge = 0.2089 lb.

Total heat units in 1 lb. of steam, at 75 lbs., produced from water at 60° F. = 1151.

Total heat units in 1 cu. ft. of steam at 75 lbs. = 1151 × 0.2089 = 240.44.

To produce by compression in a steam-actuated air compressor 1 cu. ft. of compressed air at 75 lbs. gauge and 60° F., about 2 cu. ft. of steam at the same pressure are required,† making the thermal cost of 1 cu. ft. of compressed air, at the above temperature and pressure, 240.44 × 2 = 480.88 heat units. The air is here supposed to have lost its heat of compression by being stored or transmitted to a distance, so that the 480.44 heat units represent its cost at the motor where it is to be used.

The result of reheating may now be stated:

Weight of 1 cu. ft. of compressed air at 75 lbs. and 60° F. = 0.456 lb.

Units of heat required to double the volume of 1 lb. of free air at 60° F. = 123.84.

Units of heat required to double the volume of 1 cu. ft. of compressed air at the same temperature and pressure = 123.84 × 0.456 = 56.47.

Comparing the thermal cost of 1 cu. ft. of air compressed in a cylinder with that of 1 cu. ft. obtained by reheating:

* Frank Richards, "Compressed Air," p. 158.

† That is, the efficiency of the compressor is assumed to be fifty per cent.

$$480.88 : 56.47 :: 1 : 0.1174$$

that is, the cost in heat units of the volume of air produced by reheating is less than $\frac{1}{8}$ of that required to produce the same volume by compression.

It is not to be expected that the theoretical result here set forth can be attained in practice. To effect such a saving a very perfect form of reheater would have to be employed, and the air after reheating pass directly into the cylinder of the engine. A farther conveyance of the air in pipes for even a very short distance rapidly lowers its temperature and therefore its pressure.

Temperatures Employed in Reheating. At a constant pressure the volume of air is proportional to its absolute temperature, or 459° F. plus the sensible temperature above the zero point, as read on the thermometer. The absolute temperature of air at 70° F. is $459+70=529°$. In doubling the volume by the application of heat the absolute temperature becomes 1058°, and $1058-459=599°$, which is the corresponding sensible or thermometric temperature. But this temperature is greatly reduced by the time the air reaches the motor cylinder, and still more heat is lost in the cylinder before its work is done. To reheat the air to a temperature which would really double its volume in the motor cylinder itself would involve a temperature in the reheater much higher than 599°. But such high temperatures cannot be employed, because they would render impossible the proper lubrication of the cylinder. If the temperature be raised by the reheater to 400° F. a loss of, say, 100° should be allowed for cooling before the air is actually used. The absolute cylinder temperature is then $300+459=759°$, and the corresponding added volume of compressed air practically available is found by the proportion:

$$529 : 759 :: 1 : 1.43+$$

That is, there has been an effective increase of about 43 per cent. in the volume of compressed air by heating in the reheater to 400°. It is improbable that a higher temperature would be desirable in the motor cylinder, or that any material further increase in economy could be realized in the operation of a compressed-air motor. In actual practice the gain derived from

reheating is usually considerably less than is here shown. For air engines taking air at nearly full stroke, such as machine-drills and small, single-cylinder pumps, the increase of work ranges from, say, thirty to thirty-five per cent., without deducting the cost of the fuel used in the reheater. A higher efficiency is shown for expansive-working engines.

For some purposes the determination as to the quantity of heat to be imparted in reheating is based on the temperature at which the air leaves the compressor cylinder, the idea being to recover the heat subsequently lost in cooling. Suppose, for example, that the compression is practically adiabatic, as is usually the case in single-stage dry compressors. Taking as the unit 1 lb. of air, or 13.2 cu. ft., at a temperature of 65° F., and compressing to 70 lbs. gauge, the heat of compression * is:

$$T' = T\left(\frac{P'}{P}\right)^{0.29} = 65^\circ + 459^\circ\left(\frac{70+14.7}{14.7}\right)^{0.29}$$

= 869° absolute temperature, and the final thermometric temperature is, $869^\circ - 459^\circ = 410^\circ$ F. The rise in temperature due to compression is therefore:

$$410^\circ - 65^\circ = 345^\circ \text{ F.}$$

If the compressed air be subsequently cooled to 65°, its volume becomes: $\frac{14.7 \times 13.2}{84.7} = 2.29$ cu. ft.

In using this air without reheating and non-expansively, in a machine such as a rock-drill, having, say, 10 per cent. clearance, the work done is

$$W = (2.29 \times 144 \times 84.7 \times 0.9) - (2.29 \times 144 \times 14.7) = 20290 \text{ ft. lbs.}$$

But if the air be reheated to the final temperature of compression (345° F.), the work is:

$W' = \frac{869^\circ}{524^\circ} \times 20290 = 33478$ ft. lbs., and the work gained by reheating is therefore:

$$33478 - 20290 = 13188 \text{ ft. lbs, or 39 per cent.}$$

The thermal cost of reheating this air will be: $345^\circ \times 0.2375$

* See Chapter X

(specific heat of air at constant pressure) $= 81.9$ thermal units (B. T. U.), which are equivalent to $81.9 \times 772 = 63226$ ft. lbs. of work.

Hence the efficiency of reheating in this case is:

$$\frac{13188}{63226} = 20.8 \text{ per cent.}$$

In a series of experiments carried out in connection with the large plant of the Paris Compressed-Air Company, and using an improved form of reheater, the expenditure of coke in the heater, for one added horse-power per hour, was only 0.2 pound, which is say about one-eighth of the fuel consumption of large compressors of the best make, with compound steam cylinders. But with this particular plant the above very low fuel consumption in the heater was probably greatly exceeded.

A working test, conducted by Prof. Alex. B. W. Kennedy, on a reheater supplying air for a small motor, gave the following results: The air was reheated to 315° F., with a consumption of about 0.39 lb. coke per indicated horse-power per hour, producing an increase of about 42 per cent. in the volume of the air, and, if the indicated efficiency had remained the same as during the trials with cold air, there should have been a decrease of air consumption in the ratio $\frac{1}{1.42} = 0.70$. The volume of cold air used (admission temperature, 83° F.) was 890 cu. ft. per horse-power per hour; the volume when reheated was 665 cu. ft., or 75 per cent.; so that the full economy resulting from reheating was nearly realized. In this connection Professor Kennedy says: "I do not doubt that the stoking of the heater during my experiment was much more careful than it would be in ordinary practice, although, on the other hand, it would not be difficult to design a more economical stove. If, however, the coal consumption were even doubled, it would only amount to 72 lbs. per day of 9 hours for 10 indicated horse-power, the value of which might be 6d. or 7d. The air saved per day under the same circumstances would be over 20,000 cu. ft., the cost of which, at the high rate charged in Paris, would be 7s. 3d."

A summary of the mean results obtained from two experiments on the above plant with cold, and two with reheated, air show:*

1. With cold air. Incomplete expansion, wire-drawing, and other such causes, reduced the actual indicated horse-power of the motor from 0.50 to 0.39.

2. By heating the air to about 320° F. the actual indicated horse-power at the motor was increased to 0.54. The ratio of gain due to reheating was therefore $\frac{0.54}{0.39} = 1.38$.

3. Deducting the value of about 0.39 lb. coke per indicated horse-power per hour, used in heating the air, the real indicated efficiency of the whole process becomes 0.47, instead of 0.54, and the ratio of gain is reduced to $\frac{0.47}{0.39} = 1.205$.

These carefully conducted experiments, though not made with a well-designed reheater, are valuable in proving that a substantial net gain is obtained from reheating. Where reheating is employed in mine practice, however, the quantity of heat imparted to the air is usually much less than that indicated above. Good results may be obtained by the application of even less than 100° F.

The results of some experiments by Riedler and Gutermuth, on the consumption of reheated air, by an ordinary single-cylinder eighty-horse-power engine, are given in Table XXVIII.† This

TABLE XXVIII

Test.	Temperature of Air.		Consumption Free Air per H.-P. Hr. in Cubic Feet.	Indicated Horse-Power.	Efficiency.
	Admission.	Discharge.			
1	264.2° F.	69.8° F.	462.77	72.3	0.89
2	305.6	84.4	431.09	72.3	.90
3	320.0	95.0	418.55	72.3	.91
4	338.0	120.2	432.12	65.0	.87

* "Experiments upon the Transmission of Power by Compressed Air in Paris." Van Nostrand's Science Series, No. 106, p. 35.

† Wm. Cawthorne Unwin, *ibid.*, p. 104.

engine, with Corliss valve gear, was originally designed and used as a steam engine, and no changes were made for adapting it to work with compressed air, except that the cylinder was jacketed by the hot air on its way to the valve chest. The initial pressure was 95.5 lbs. absolute and the temperature of the air in the reheater did not exceed 338° F., at a coke consumption of 0.176 lb. per horse-power hour.

Construction and Operation of Reheaters. The reheater employed in the experiments referred to in the preceding section was that in use some years ago in connection with the Paris plant. It consisted of a double cylindrical box of cast-iron twenty-one inches diameter by thirty-three inches high, over all, enclosed in a

Fig. 124.—Leyner Compressed Air Reheater.

sheet-iron casing. The air under pressure traversed the annular space between the inner and outer cylinders, being compelled by baffle-plates to circulate in such a manner as to come into contact with the whole heating surface. The products of combustion, from a coke fire in the inner cylinder, passed downward over the

exterior surface of the annular air chamber on their way to the chimney. A heater of this size served for a ten to twelve horse-power motor.

In another form of reheater the air is passed through a coil of wrought-iron pipe, enclosed in a cylindrical casing. A large heating surface is thus obtained, but wrought-iron pipe is objectionable because it burns out rapidly unless the fire is kept moderate. The conditions are materially different from those to which boiler tubes are subjected, since the air tubing is denied the cooling effect of the water. Cast-iron coils, on the contrary, such as those of the Leyner reheater (Figs. 124 and 125), stand well. The U-shaped pipes are made in separate sections, bolted together as shown, with asbestos packing in the joints. By varying the number of units any desired capacity can be obtained, and a broken or burned-out section is readily replaced.

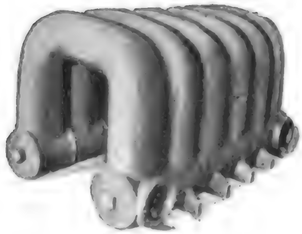


FIG. 125.—Cast-Iron Coils, Leyner Reheater.

The Sergeant reheater (Fig. 126) consists of two concentric cast-iron shells, bolted together, one within the other, the joints being packed with asbestos gaskets. The inner chamber forms the top of the fire-box. In shape this reheater is a truncated cone, set on a cylindrical fire-box, the cold-air main being connected by a flange coupling at the top and the hot air discharged near the base. This heater measures 42 ins. outside diameter at the base by 54 ins. high, with a grate 19 ins. diameter. It is stated that 340 cu. ft. of free air per minute, at 40 lbs. pressure, can be heated to 360° F., with a gain of 30 to 35 per cent. in the energy developed. If more than 400 cu. ft. of free air per minute are to be reheated, 2 or more heaters of this size should be set in series, the air passing from one to another, allowing a maximum of 400 cu. ft. for each.

The inner and outer shells of reheaters of the cast-iron-shell type are subjected to considerable differences of temper-

ature, and except when of small size the upper and lower ring joints between the shells are difficult to keep tight.* In the Rand reheater (Fig. 127) the castings are more complicated in shape, the air passing between them in a thin sheet, from the inlet on the side to the discharge at the top of the central dome. To provide for expansion and contraction, the lower joint above the

FIG. 126.—Sergeant Reheater.

fire-box is provided with a stuffing ring and packing, shown in the cut. There is still a tendency to leakage, however, if the fire be very hot.

The Sullivan reheater (Fig. 128) is quite different in design from those described above, consisting essentially of a vertical coil of cast-iron piping, or hollow rings, encased in double sheet-steel shells, the space between the latter being filled with asbestos.

* *Sibley Journal of Engineering*, 1904.

Below is the grate and combustion chamber, the gases from which pass through the spaces between the air rings. To minimize leakage, the centers of the rings are joined by malleable-iron nipples, so that all expand and contract together. These heaters,

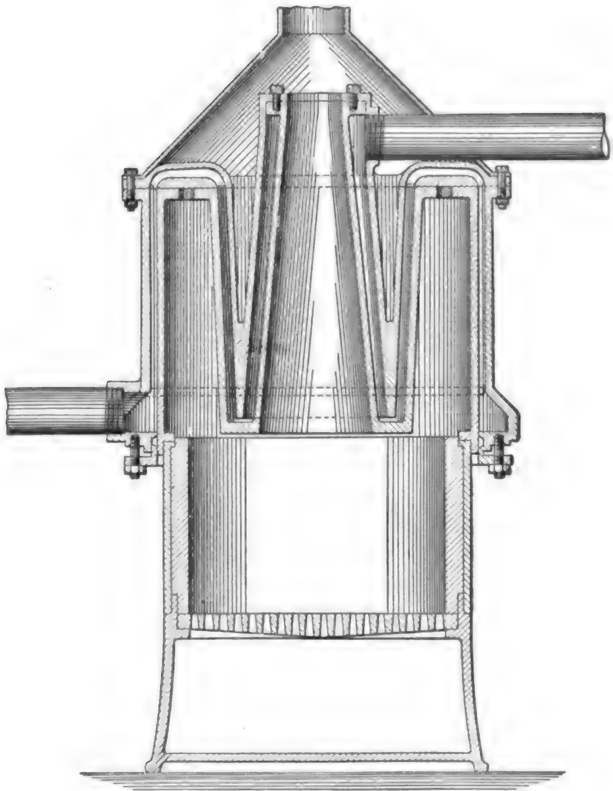


FIG. 127.—Rand Reheater.

usually designed for burning coal, coke, or wood, are made in 3 sizes, for 345 to 800 cu. ft. of free air per minute, having from 3 to 7 rings, and measuring from 5 ft. 8 ins. to 7 ft. 6 ins. in height, by 33 ins. outside diameter.

Internally fired reheaters—those in which the air is heated by direct contact with the fire—have hitherto been unsuccessful,

because dust and injurious products of combustion are carried by the air into the cylinder of the air motor. This trouble, of course, does not exist to the same extent when gasoline or other oils are used, instead of solid fuels, nor in the electric reheater, which, however, has thus far had but a limited application.

A fault of most reheaters as built at present is that there is no provision for regulating the heat according to the variation in consumption of air, such as is unavoidable in applying reheating for machine drills, channellers in quarry work, hoisting engines, and other intermittently operating machinery. This want of regulation evidently is not so important for constant-running engines, such as pumps.

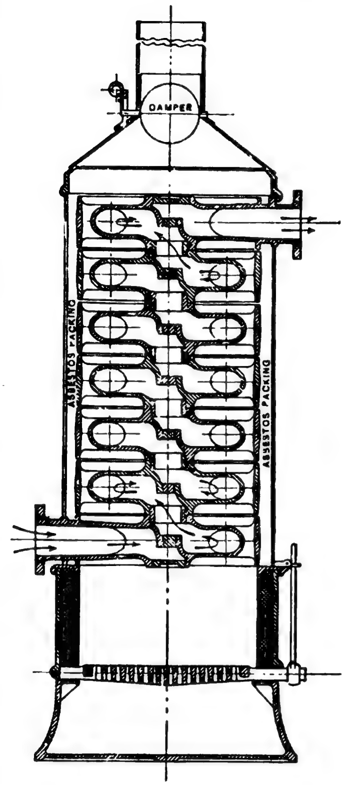


FIG. 128.—Sullivan Reheater.

As the air chamber, of whatever shape, in all of the externally heated or "dry" reheaters, forms in reality a part of the air main, reheating can increase the *pressure* only in a small degree. Its real effect is to increase the *volume* of air, which tends to back up in the main, reducing incidentally the velocity of flow and therefore the loss of pressure due to friction. The reheater should always be placed as close as possible to the machine using the air. This is readily done with stationary engines, like pumps or hoisting engines; and even in the case of movable machines, like quarry channellers, the reheater may be set on the same carriage or bed-frame. If it be necessary, however, to convey the heated air some distance, the temperature may be quite effectually

maintained by covering the pipe with non-conducting material, as is done with steam piping.

Sometimes when the air-engine cylinders are compounded, the exhaust from the high-pressure cylinder is passed through a second reheater before going to the low-pressure cylinder. A further benefit may be derived by injecting into the reheater a very small quantity of water. The specific heat of water is high as compared with the specific heat of air; also such part as is converted into steam gives up its latent heat in the motor-engine cylinder and prevents trouble from freezing, even when a high rate of expansion is employed. For the same reason, benefit may be derived from injecting a little warm, or even cold, water into the compressed-air feed-pipe of an air motor. Water used in this way acts incidentally as a mechanical scourer, in washing away accumulations of ice tending to form in the ports.

It will be seen from the construction of reheaters that the calorific power of the fuel burned in them is not economically utilized. The flue loss is large for the same reasons that apply to the work of ordinary shell or tubular boilers. But the thermodynamic advantage gained is so considerable that the low efficiency of the reheater itself, in burning the small quantity of fuel required, becomes of secondary importance.

Use of Reheaters for Underground Work. In the ordinary operations of mining the reheating of compressed air can have only a limited application. By far the most important use of compressed air in mining is for operating machine drills. Up to the present time there are relatively few mines where it is employed for any other purpose. But it is evident that for portable machines like rock-drills, continually being shifted from place to place underground, the use of reheaters in most cases is economically out of the question. Not only would a number of them be necessary, but they would have to be moved about and kept close to the drills, to prevent the reheated air from losing its heat and temporary increase of volume.

For stationary engines, however, such as underground pumps, hoists, rope-haulage engines, etc., and wherever the reheater can

be placed permanently close to the air engine, reheating in mines may be successfully applied. The idea that it is useful mainly in preventing the accumulation of ice in the exhaust ports and passages of the air engine is not an uncommon one; but as a matter of fact the prevention of freezing is merely incidental to a decided gain in efficiency. In underground work it may be difficult to arrange for burning fuel under a reheater, notwithstanding the small quantity required, because of the resulting vitiation of the mine atmosphere. Also, in gassy collieries reheaters cannot well be used, though sometimes the products of combustion may be turned into an upcast airway, or even allowed to escape into the mine workings, when the heater is small and the active circulation of large volumes of air is maintained. Where the conditions underground are such that strong combustion is not allowable and only a small quantity of fuel can be burned in the reheater, it will still be found that some advantage is obtainable from air engines by a very slight added temperature—say, only 25° to 50° F. In this connection it may be noted that the use of the internal electric reheater, already referred to, in which a resistance coil is placed in an enlarged section of the air main, does away with the difficulty of disposing of the products of combustion of fuel and would be especially useful in gassy collieries. Another mode of applying electric reheating is to wrap the resistance coils around a short length of the air pipe.

At the North Star Mine, Grass Valley, Cal., the plan has been adopted of placing a reheater on the surface near the shaft mouth and carrying the compressed air underground by a pipe covered with non-conducting material. Fairly satisfactory results are thus obtainable, with the advantage of avoiding the burning of fuel in the mine. But while some saving can be realized in this way for moderate distances—say of a few hundred feet—it would be economically out of the question for long transmission lines. This arrangement suggests the caution that non-conducting covering should always be used for the pipe from reheater to air engine, however short the distance. In a case on record,* where

* Richards, *American Machinist*, Feb. 28th, 1895.

this distance was only 20 ft., but no pipe covering provided, the gain in power realized was only 12 per cent., though the absolute temperature of the air was increased at the reheater 38 per cent., with of course the same theoretical increase of volume. The air used for operating an underground pump at another California mine is reheated by steam conveyed from the surface. Steam may thus be used to greater advantage than if employed directly in the cylinder of a pump; for, in condensing, the latent heat otherwise lost is utilized in raising the temperature of the air and is so converted into work. All devices of this kind, however, must be classed as makeshifts.

In recent years several mine plants have been erected at which compressed air has been used even for operating surface hoisting engines—for example, at the Lightner Mine, Calaveras Co., Cal. One of the units of a battery of boilers is adapted as a reheater. The compressed air passes from the receiver into a section of perforated pipe submerged just below the surface of the hot water in the boiler, and is thence led to the hoisting engine. By means of a large receiver capacity, quite satisfactory results are secured, notwithstanding the intermittent work of the engine.

In connection with the method of reheating referred to above the results may be given of a number of experiments made by Prof. J. T. Nicholson, in reheating air from the Taylor Hydraulic Air Compressor, at Magog, Prov. Quebec, described in Chapter XV. The air was used in a 27-horse-power Corliss engine, at a pressure of 53 lbs. There were 5 tests, as follows: 1. With cold air. 2. Reheating by means of steam injected into the air main before reaching the engine. 3. The compressed air was passed into a steam boiler, and heated by mixing with the water and steam. 4. The compressed air was blown upon the surface of the water in the boiler, and heated by mixing with the steam. 5. The air was passed through a tubular reheater, fired by coke.

Without reheating, 850 cu. ft. of free air were used per indicated horse-power hour. By reheating in the boiler, a mixture of 10 to 15 lbs. of steam with the air reduced the consumption of air from 850 cu. ft. to 300 to 500 cu. ft., per indicated

horse-power hour. Thus, 1 added horse-power was obtained by *wet heating*, at an expenditure of from 1 to 1.3 lbs. of coal per horse-power hour.

By *dry heating* in the coke-fired reheater, the air was raised to 287° F. At this temperature, 640 cu. ft. of free air were required per horse-power hour, or 210 cu. ft. less than with cold air, the saving in the quantity of air being about 25 per cent. By burning in the reheater 47 lbs. coke per hour, 100 horse-power in cold compressed air was raised to 133 horse-power, making an expenditure of 1.42 lbs. coke per hour for each added horse-power. These results indicate that the reheater used was not very efficient. But though the fuel consumption was much greater than in Professor Kennedy's test, previously described, it is still far lower than is attainable in the most efficient engine and boiler practice.

In a paper by Clarence R. Weymouth, on "Reheating Compressed Air with Steam,"* a detailed discussion is given of the thermodynamics of this mode of procedure, with deductions as to its efficiency. The author considers the cases of injecting steam into the air main, and of passing the compressed air through a steam boiler, giving the results in tabulated form.

* "Compressed Air Information." Edited by W. L. Saunders.

CHAPTER XX

COMPRESSED AIR ROCK DRILLS

In the introductory chapter reference was made to a few of the facts relating to the earlier stages of development of the modern rock drills, and to the importance of these machines in the working of mines and quarries, the sinking of shafts and the driving of tunnels. The machine drill has not only been the means of increasing greatly the speed of work, thereby reducing the cost of all operations involving rock excavation, but it has made possible the driving of long tunnels, which, it is safe to say, could never have been completed by hand drilling.

It is unnecessary here to trace the history of machine drills. This has been well done in Drinker's treatise on "Tunneling, Explosive Compounds, and Rock Drills," first published in 1878. In that book details are given of many of the numerous patents taken out in the United States and Europe, from 1849 to 1882, and the successive steps in the earlier development of the machine drill are fully recorded. The present chapter will be devoted chiefly to the description of a number of representative drills now in use, together with notes on the performance, consumption of air and other matters relative to the operation of machine drills. While it may not be said that rock drills have become so standardized in design that further important improvements are improbable, yet, in the past eighteen or twenty years, and especially since the "air hammer" drill was introduced, radical changes in principle have been few. Weak points in design and construction have been discovered by experience, and remedied as far as practicable, so that at the present time there are a number of successful, serviceable machines on the market.

For rock work the percussion drill only is of practical use, at least for any rock harder than soft shales, coal, and other similar

material. All attempts to construct a rotary pneumatic rock drill have thus far failed. The diamond, and other core, drills for deep boring, and the efficient Brandt rotary drill, operated by hydraulic power, obviously have no place in the present discussion.

General Description. The reciprocating or percussion rock drill, aside from those machines that operate on the hammer principle (see Chap. XXI.), may be roughly described as consisting of a cylinder, in which either compressed air or steam is used, the drill bit being firmly clamped to a chuck, forming the end of the piston rod (Fig. 129.) For admitting and exhausting the compressed air alternately at each end of the cylinder a number of different valve motions are in use. Some of these are similar to the valve motions of certain single-cylinder pumps; in others a positive movement of the valve is caused by the introduction of a tappet, actuated by the strokes of the piston. There is also a device to produce automatically the necessary rotation of the drill bit on its axis, for keeping the hole round and preventing the bit from sticking. In standard drills this is done by a rifle bar, ratchet and pawls. Working speeds are usually from 300 to 400 strokes per minute, for the larger sizes of drills, up to 500 strokes for the smaller; the normal length of stroke, in drills of average size, being from 4½ to 6 inches. The admission of air, and therefore the speed and force of the blow delivered, are controlled by a hand valve in the air pipe close to the valve chest.

A feed screw, with crank and handle, is carried in a bearing at the rear end of the shell supporting the cylinder, and engages with a nut on the under side of the cylinder casting. By this means, the entire drill head is fed forward by hand as the hole is deepened, several bits of successively greater and greater length being clamped as required in the piston rod chuck. (An automatic feed has been introduced, and used to some extent for surface work, but is neither necessary nor entirely satisfactory for underground service.) When the cylinder has been fed forward as far as the length of the screw and of the shell will permit, the drill is stopped. By reversing the feed the cylinder is moved back on its supporting shell, the bit removed and a longer bit put in the chuck. The cylinder

is then fed forward until the new bit touches the bottom of the hole, with the piston nearly at mid-stroke; the air is turned on slowly and the work proceeds. It will be seen from this that the length of stroke, and therefore the force of the blow, are under the control of the drill-runner. If, while the machine is at work, the cylinder is fed forward faster than the hole is being deepened the stroke necessarily becomes shorter and shorter, because the bit strikes the bottom of the hole before the full length of stroke is reached; on the other hand, should the feed be too slow, the piston will strike the front cylinder head. Thus, the force of the blow may be graduated to suit the conditions. When starting a hole, for example, the stroke should be shorter than when some depth has been reached and the bit has adjusted itself to the shape of the bottom of the hole. Moreover, for hard rock, a short, rapid stroke gives the best results; while a longer stroke may be adopted for softer or tough rock.

Further details of the construction and operation of machine drills are given hereafter, in describing examples of the different makes.

Modes of Mounting Drills. The drill head, comprising the cylinder and its appurtenances and the supporting shell, may be mounted either on a tripod or column. For surface work the tripod only can be used; underground, either the tripod or column, depending on the size and shape of the working in which the drilling is to be done.

1. *Tripod.* (Fig. 129.) The legs, which are telescopic, are hinged by a heavy bolt to the tripod head, and can thus be set as necessary for adjusting the position of the axis of the cylinder and bit, for the hole to be drilled. To the tripod head is bolted the "shell," which is provided with guides for supporting the cylinder, as it is fed forward. After the machine has been placed in position for drilling all bolts are tightened up. Heavy weights are usually slipped on the tripod legs, to prevent the drill from shifting while in operation.

Tripod mountings are required not only for surface drilling, but also for underground work, when the distance between roof

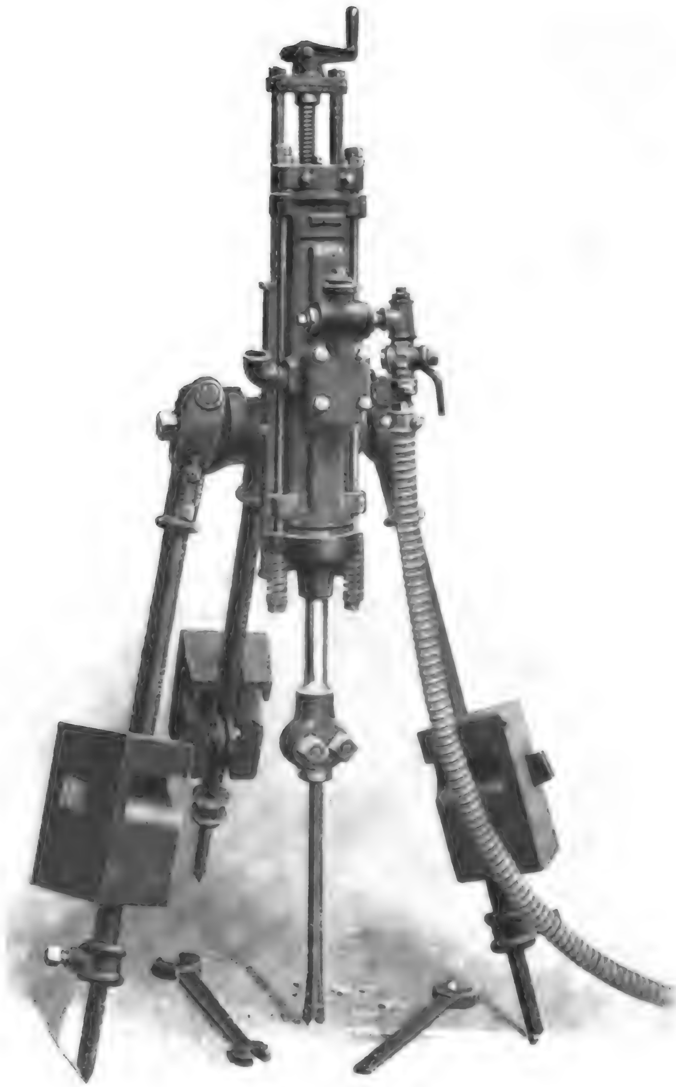


FIG. 129.—Sullivan Tappet Drill.

and floor, or between the side walls of the workings, is too great to permit the use of columns. Where a choice exists, however, the tripod is sometimes employed in preference to the column; because, as a rule, it may be set up with less loss of time and allows

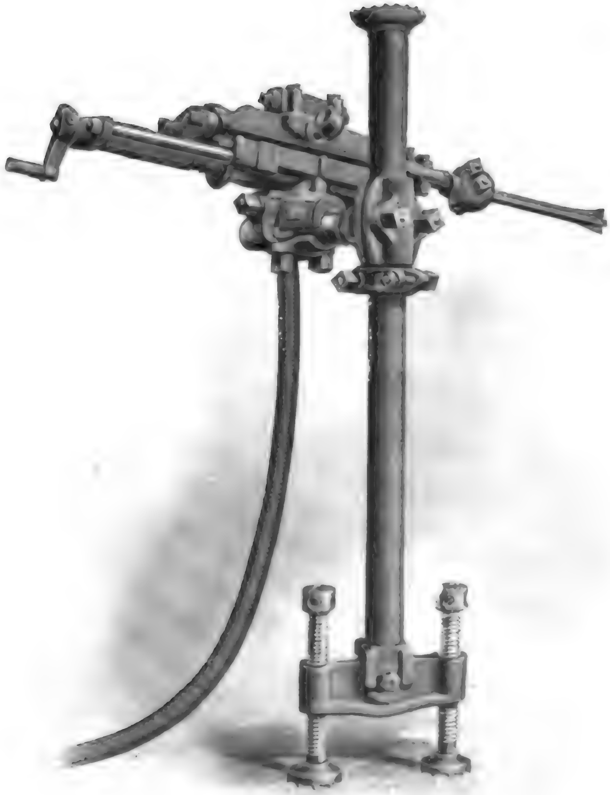


FIG. 130.—Double-Screw Column Mounting for Rock Drills.

greater freedom in locating the holes, to produce the best results under existing conditions as to character of rock and shape of the face. The same may be true, also, in sinking shafts of large cross-sectional area, or when the rock is so irregularly fissured that a symmetrical arrangement of the holes is undesirable.

2. *Column or Bar.* (Fig. 130.) The cut shows the usual form; a hollow, iron bar, varying in diameter from 3 in. to 5½ in., according to the size and weight of the drill. It can be used only underground and is securely set between the roof and floor, or the walls, of the working. The lower end of the column has a cross-piece or base, through which pass a pair of jack-screws. The upper end terminates in a serrated cap or head, which, by tightening up the screws, takes a firm hold upon the surface against which it is pressed. Another form of column has a single, telescopic jack-screw; it may be used in small tunnels or mine workings, and also for shaft-sinking. In mine workings of large size the double-screw column is preferable, the drill being carried on an arm, attached to a collar encircling the column. When necessary two drills are mounted on the same column. The collar, or collar and arm, slides on the column longitudinally, and may also be revolved around the column. It is clamped fast in any position, as desired, for adjusting the height and angular direction of the drill. For the single-screw column the drill is attached directly to the collar, the arm being omitted.

The column mounting is specially useful in driving mine tunnels, cross-cuts, and drifts, and for the advance headings of railroad tunnels. It is frequently employed, also, for stoping, when the pitch of the vein and the method of mining make it inconvenient to use tripods. When placed in an approximately horizontal position, as in shaft-sinking, the column is known as a "bar," though the mode of mounting the drill upon it is substantially the same. Shaft-bars are sometimes made extra long, for wide shafts, in which case a pair of adjustable legs are hinged to a collar at the middle point to carry the weight of the drill or drills without making the bar inconveniently heavy.

Formerly, for driving railroad tunnel headings, four, six or more, machine drills were mounted on a carriage, running on rails like a car, but these are no longer used in general practice.

Classification of Compressed Air Drills. There are two distinct classes: *First,* Reciprocating drills, a name which may be given to those in which the drill bit is firmly attached to the piston rod, and

delivers a succession of blows on the bottom of the hole; *second*, Air Hammer drills, in which the bit does not reciprocate, but is held in the forward end of the cylinder, and is struck by the piston as by a hammer.*

RECIPROCATING DRILLS.

These constitute the more important of the two classes and comprise a number of types and makes. Among the best known American drills are the Doble, Ingersoll, Jeffrey, McKiernan, Murphy, Rand, Sergeant, Sierra (Rix), Sullivan, Temple "Air-Electric," and Wood. Some of these are widely used throughout the world. Of the European reciprocating drills the following may be mentioned: Adelaide, Barrow, Chersen, Climax, Darlington, Ferroux, Froelich, Hirnant, Holman, "Konomax," Küzel, Little Wonder, Meyer, Rio-Tinto, Schram, Triumph, and "Währwolf."

Based on the design of the valve-motion, reciprocating drills may be further classified as: tappet-valve and spool- or piston-valve machines, and those in which the valve is eliminated completely, its function of controlling the admission and exhaust of the compressed air being exercised by the piston itself. Examples of each form are given in the following pages.

"Sergeant" Rock Drill. Fig. 131 shows a longitudinal section of this machine, which is built by the Ingersoll-Rand Company. The spool-valve and the main air and exhaust ports are similar in some respects to those of a single-cylinder pump. Air is admitted on one side of the valve chest, the exhaust opening being on the other side.

The valve-motion is non-positive and consists of two parts: a spool-valve, which controls the main cylinder ports and an arc-shaped tappet, set in a correspondingly curved slot or seat, as shown, between the cylinder and valve chest. The ends of this tappet, which project slightly into the main cylinder, are struck alternately by the front and back shoulders of the large annular

* Air Hammer drills are discussed in Chapter XXI.

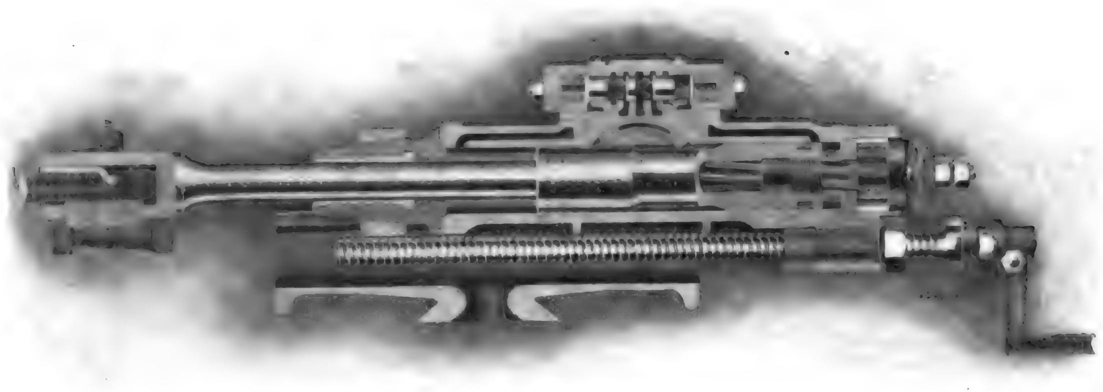


FIG. 131.—"Sergeant" Rock Drill. Ingersoll-Rand Co.

recess in the middle of the piston, thus causing the tappet to oscillate at each stroke of the piston. Behind the tappet, and in the vertical wall of its seat, are three small auxiliary ports, one in the middle and one near each end of the seat. These auxiliary ports connect with the spool-valve chest above; the middle port with the middle of the chest, the rear port (*i.e.*, nearest the back end of the cylinder), with the *forward end* of the chest and the forward port with the *rear end* of the chest. In the face of the tappet is a curved slot, just long enough, when in the extreme positions of its throw, to form a communication between the middle auxiliary port and one of the end auxiliary ports. That is, at each stroke of the main piston, the tappet places the *opposite end* of the valve chest in communication with the exhaust, thus causing the throw of the spool-valve. In the peripheral surface of the spool-valve a very fine longitudinal slot is cut, which constantly admits a small quantity of live air to both ends of the chest. Hence, when either end of the chest is connected with the exhaust, as stated above, the valve is thrown towards that end by the air pressure in the other end of the chest.

In Fig. 131, the piston is beginning its forward stroke. The spool-valve, in its rear position, is admitting air to the back end of the cylinder, while the forward end of the cylinder is connected with the exhaust. As the piston advances, the rear shoulder of the annular recess in the piston strikes the projecting end of the tappet and throws it over. This changes the relation between the auxiliary ports, already described, exhausts the air from the front end of the chest and throws the spool-valve forward, thus preparing for the back stroke of the piston.

By the introduction of the arc-tappet, the Sergeant drill avoids in large part one of the chief defects of the ordinary spool-valve drills, *viz.*: irregularities in the operation of the machine, caused by wear of the valve and seat, which permits leakage of the air or steam past the valve. Adjustments for any irregularities of stroke produced by wear are made by the simple compensating device shown in Fig. 132, which is an enlarged section of the valve and chest. A hollow brass plug, *P*, having a very small hole, *H*,

permits the passage of a little live air to the back end of the chest. Should the piston strike the back cylinder head, the area of H is reduced slightly by peening or riveting up the edge of the hole. This decreases the quantity of air passing to the end of the chest and increases the cushioning in the rear end of the cylinder. If the stroke be too short, H may be found partly obstructed and should be cleaned, to admit more air to the end of the chest; if the stroke be still too short, the area of H is slightly increased with the point of a knife blade.

The rotation of the piston and bit is caused as follows: A rifled bar, with a ratchet head and pawls set in the rear cylinder

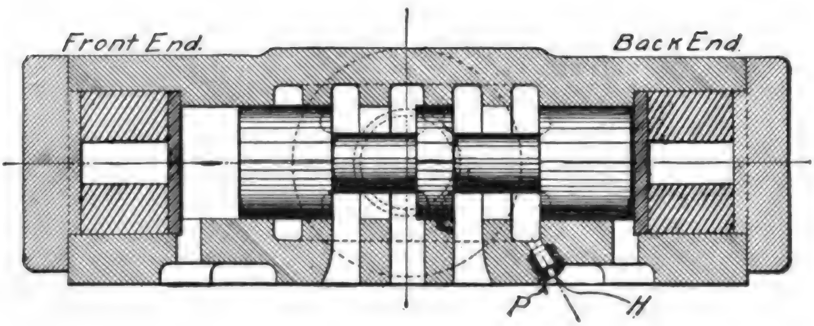


FIG. 132.—Spool-Valve and Chest. "Sergeant" Rock Drill.

head, engages with a correspondingly rifled nut screwed into the end of the hollow piston. The teeth of the ratchet wheel and its pawls are so shaped that, on the forward stroke, the piston moves without rotation, the rifle-bar turning the ratchet in its seat. On the back stroke the pawls prevent the ratchet from turning, so that the piston is compelled, by the rifle-bar and nut, to rotate through a part of a revolution. The ratchet ring, with internal teeth, with which the pawls engage, is not fastened rigidly in the back cylinder head, but is held by friction only, under pressure of an exterior cushion spring, acting on the periphery of the ratchet. Hence, when the drill bit sticks in the hole, or for any reason cannot rotate freely on the back stroke, the ratchet itself turns, thus preventing injury. The drill is fed forward on its supporting shell by a long

feed screw, engaging with a feed nut in a lug on the under side of the cylinder.

The Sergeant drill is built in seven sizes, the cylinders measuring: 2 in., $2\frac{1}{4}$ in., $2\frac{1}{2}$ in., $2\frac{3}{4}$ in., 3 in., $3\frac{1}{4}$ in., and $3\frac{1}{2}$ in. diameter; weights of drill-head, unmounted, range from 110 to 405 lbs.

Sullivan "Differential" Drill. Fig. 133 shows this drill as designed specially for steam. While the design and construction are essentially the same, the spool-valve of the air drill is ground with a larger clearance, to reduce the danger of freezing when the air is exhausted. Also, the front head is modified. Instead of the soft, steam packing 29, and the gland 28, a metal lining is used, with a leather packing ring. Several types of air head are designed for this machine, a bushing being sometimes provided. Fig. 134 shows this drill as designed for using compressed air.

Referring to Fig. 133, the chest 2 contains the spool-valve 6, air inlet port 3, exhaust ports 4, 4, and cylinder ports 35, 35. The valve is thrown by the action of the small reverse ports 5, 5, communicating from the ends of the chest to the opposite ends of the cylinder as indicated. In the cut, the piston is shown on its forward stroke. When its rear end passes the opening into the cylinder of the right-hand reverse port 5, live air is admitted to the opposite, or left-hand, end of the valve chest; thus throwing the valve to the right, to prepare for the back stroke of the piston. During the movements of the valve, the cylinder ports 35 are alternately placed in communication with the exhaust ports 4, 4 and the air inlet 3.

Rotation of the piston and bit, on the back stroke, is produced in the usual way by the rifle-bar 13 and the ratchet head 14. The ratchet has broad teeth with rounded surfaces, and steel pins or rollers are provided, instead of ordinary pawls. To prevent injury, in case the bit should become wedged in the hole and so resist rotation, the ratchet ring, as usual, is held in its seat by friction only. In addition to the piston rings 34, the piston carries two collars 36, provided with longitudinal slots, for obtaining efficient circulation of the lubricating oil. Oil is admitted from

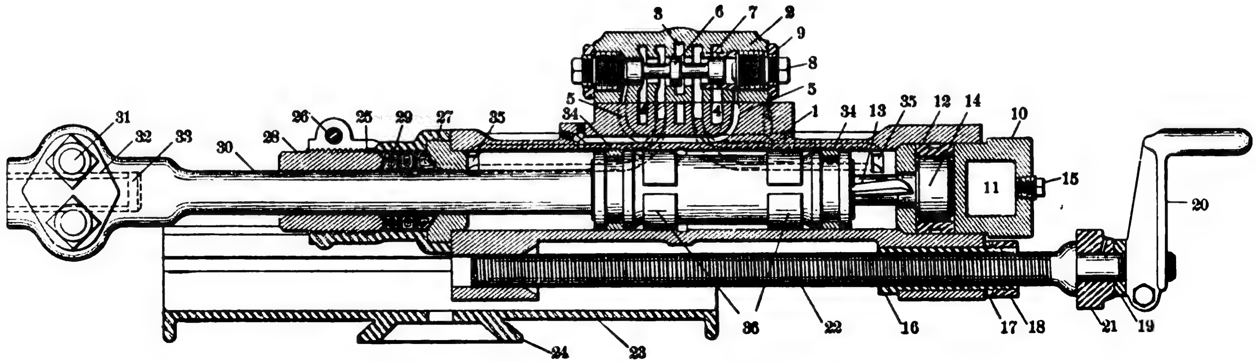


FIG. 133.—Sullivan "Differential" Drill (for Steam).

1. Cylinder.
2. Chest.
3. Inlet port.
4. Exhaust ports.
5. Reverse ports.
6. Valve.
7. Valve bushing.
8. Buffer.
9. Check nut.
10. Back head.
11. Oil chamber.
12. Ratchet ring.
13. Rifle bar.
14. Ratchet.
15. Plug.
16. Feed nut.
17. Lock washer.
18. Check nut.
19. Washer.
20. Feed handle.
21. Yoke.
22. Feed screw.
23. Shell.
24. Trunnion for mounting.
25. Forward head.
26. Clamp bolt.
27. Bushing.
28. Gland.
29. Packing.
30. Piston.
31. Clamp bolt.
32. Chuck bushing.
33. Chuck button.
34. Piston rings.
35. Cylinder ports.
36. Piston collars.

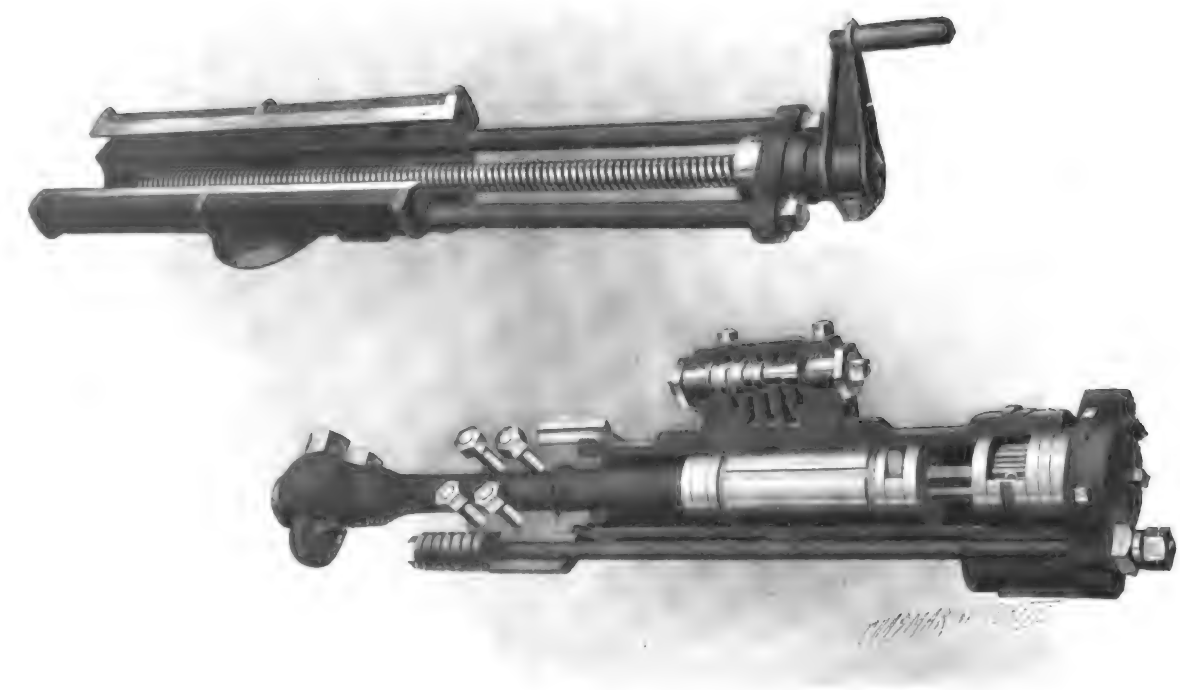


FIG. 134.—Sullivan "Differential" Drill (for Air).

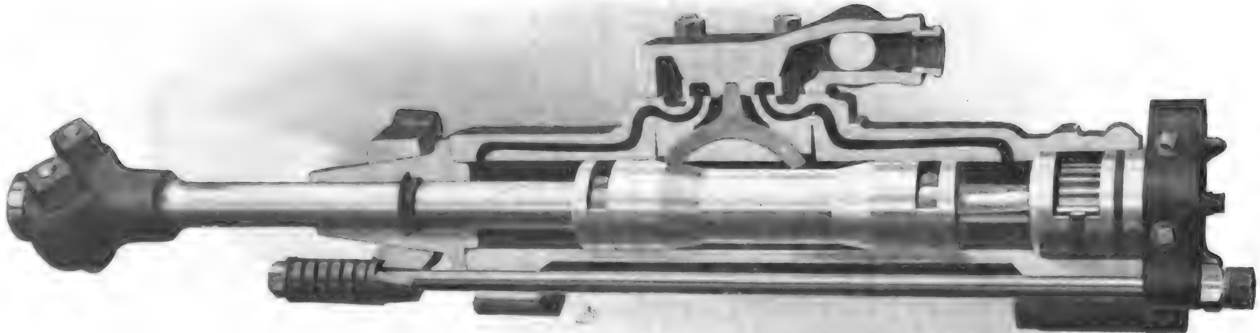


FIG. 135.—Sullivan Tappet Drill.

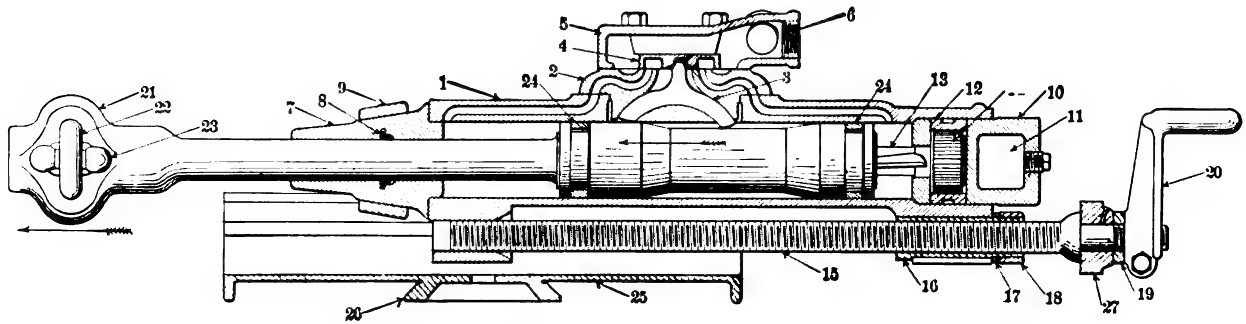


FIG. 136.—Sullivan Tappet Drill.

1. Cylinder.
2. Valve plates.
3. Rocker.
4. Valve.
5. Air chest.
6. Air inlet.
7. Front head.
8. Packing leather.
9. Ring.
10. Back head.
11. Oil chamber.
12. Ratchet ring.
13. Rifle bar.
14. Ratchet.
15. Feed screw.
16. Feed nut.
17. Lock washer.
18. Check nut.
19. Washer.
20. Feed handle.
21. Chuck.
22. Chuck bolt.
23. Clamp blocks.
24. Piston rings.
25. Shell.
26. Trunnion, for mounting.
27. Yoke.

the chamber 11, through a small passage, to the ratchet-head and thence to the cylinder itself.

The "differential" (spool-valve) drill is made in ten sizes, with cylinder diameters of: 2 in., $2\frac{1}{4}$ in., $2\frac{1}{2}$ in., $2\frac{3}{4}$ in., 3 in., $3\frac{1}{8}$ in., $3\frac{1}{4}$ in., $3\frac{5}{8}$ in., $4\frac{1}{4}$ in., and 5 in. Weight, unmounted, from 110 to 680 lbs.

Sullivan Tappet Drill. (Figs. 135 and 136.) In its general lines this is similar to the spool-valve drill of the same makers. Air is admitted (Fig. 136) at the connection 6 to the chest 5, which contains the slide valve 4, controlling the air and exhaust ports. The tool-steel rocker or tappet 3 is not pivoted, but oscillates in an arc-shaped slot, as it is struck alternately by the two beveled shoulders in the middle part of the piston. On the back of the rocker is a lug, of standard rack-tooth form, which engages with a corresponding socket in the underside of the valve 4, thus throwing the latter. Between the cylinder casting and the valve-chest is the valve plate 2, which may readily be removed for obtaining access to the rocker.

The rotation and feed mechanism are the same as in the "differential" drill. The band 9, around the front cylinder head 7, is provided with lugs for the side bolts, which run back to the rear head. On each bolt, forward of the band, is a heavy spiral spring, to take up the shock in case the piston should strike the head. In the cylinder head is a leather packing-ring 8.

The Sullivan tappet drill is made in four sizes, whose cylinder diameters are: $2\frac{3}{4}$ in., $3\frac{1}{8}$ in., $3\frac{1}{4}$ in., and $3\frac{5}{8}$ in.; weights, unmounted, from 233 to 393 lbs.

Jeffrey "Badger" Drill. Originally, a machine called the "Badger" was made by the Phillips Rock Drill Co. In an improved form it is now built by the Jeffrey Manufacturing Co. Fig. 137 is a general longitudinal section. Though of the spool-valve type, it differs materially in some of the features of its valve motion from the drills of the same class already described.

The valve *a* is a double, balanced spool, whose front end is of larger diameter than the rear. The air inlet is on the side of the machine, just below the chest, the exhaust being on the up-

per side of the chest itself. Fig. 138 is a diagram showing, in a necessarily distorted manner, the relations and connections of the system of main and auxiliary ports, which cannot be completely represented in the longitudinal section. At the beginning of the outward stroke, air enters through the port *b*, to the annular recess in the main piston; passing thence, by the lower port *c*, to the rear end of the cylinder. Simultaneously, the forward end of the cylinder and the rear end of the valve chest are exhausting respectively through the main port *d*, and the auxiliary ports, *e* and *f*. The valve is held in the position shown by the air pressure in the forward end of the chest, previously admitted through the auxiliary port *g*. When the piston has advanced a short distance, it closes the port *c* and opens the upper main port *h*. In this position of the piston air is admitted to the rear end of the cylinder through ports *b* and *i* to the valve, and thence, as indicated by the arrows, through

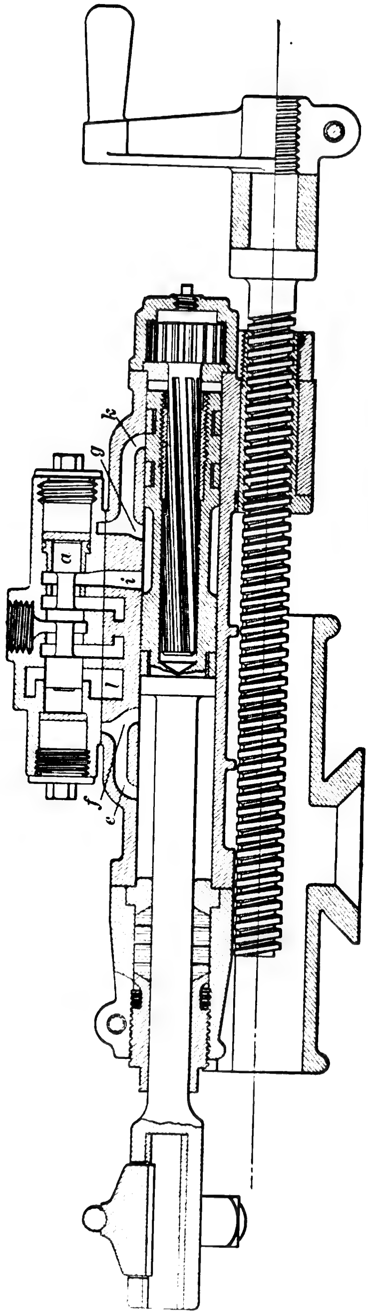


FIG. 137.—Jeffrey "Badger" Drill.

port *h*. Advancing still farther, the piston closes communication between the air inlet and the rear of the cylinder, by covering the port *i*, so that the stroke is finished by expansion of the air already admitted.

At this point, the annular recess of the piston begins to uncover the auxiliary port *f*, which admits air to the rear end of the valve chest. The valve now reverses by the difference of pressure of the live air on the small end, and of the expanded air from the cylinder (through port *g*) on the forward or large end, of the valve. This reversal prepares for the back stroke of the piston

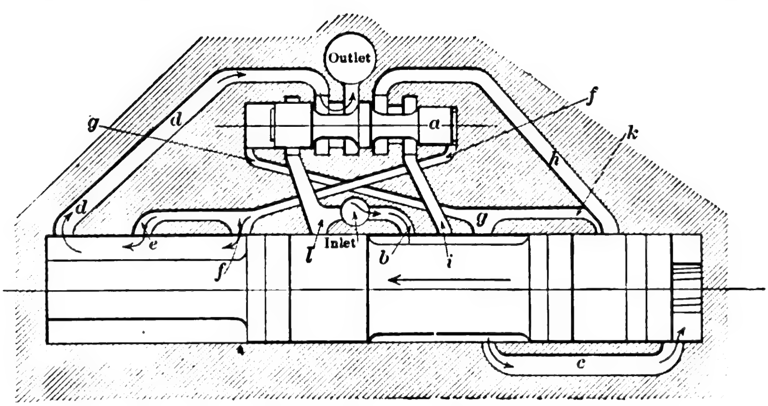


FIG. 138.—Jeffrey "Badger" Drill. Diagram of Valve and Ports.

by allowing air to enter by the auxiliary port *l*, to the chest and thence through the main port *d*, to the forward end of the cylinder. Toward the end of the back stroke, the rear ports *h* and *k* are closed by the piston and the back stroke is thus cushioned. The forward stroke is uncushioned.

The Badger drill is made in four sizes, the diameters of cylinder being: $2\frac{3}{8}$ in., $2\frac{7}{8}$ in., $3\frac{1}{8}$ in., and $3\frac{5}{8}$ in.; maximum length of stroke, from 5 in. to $7\frac{1}{4}$ in. Weight of drill, unmounted, ranges from 150 to 320 lbs.

Ingersoll-Rand "Arc-Valve" Tappet-Drill. (Fig. 139.) This machine furnishes an example of a positive valve-motion and

constitutes an improvement on the earlier tappet drills of the same makers, which have a straight-face slide valve.

The valve motion is as follows: A three-armed rocking tappet is pivoted in a recess in the upper part of the cylinder. When the drill is at work, the ends of the two lower arms of the tappet are struck alternately by the front and back shoulders of a wide, annular recess in the piston, thus causing the tappet to oscillate on its fixed center. The third arm, projecting upward, engages with a socket in the back of the slide-valve, and throws the valve positively, in an arc-shaped path, at each stroke of the piston. The seating surface of the valve, instead of being plane, is therefore also circular, its center being the center of the tappet pin. No auxiliary ports are required in this simple design; the valve controls the two main cylinder ports and the exhaust, as shown. In addition to the usual cushioning on the back stroke, common in all rock drills, the forward stroke is also slightly cushioned; this result being produced by the shape of the lower arms of the tappet, the recess in the piston and position of the main ports. The machine is provided with the usual feed and rotation mechanism.

The drill may be operated by either compressed air or steam. It is found to be better adapted to steam than the piston valve drills, which do not work satisfactorily with wet steam, or in the presence of water of condensation.

The "Arc-Valve" tappet drill is made in six sizes, the cylinders measuring $2\frac{1}{4}$ in., $2\frac{1}{2}$ in., $2\frac{3}{4}$ in., $3\frac{1}{8}$ in., $3\frac{1}{4}$ in., and $3\frac{5}{8}$ in. diameter; weights of the drill head, unmounted, are from 140 to 415 lbs.

Murphy "Little Champion" Drill. Fig. 140 shows the general plan, and longitudinal and cross-sections, of this machine, which has been on the market about ten years. The rotation mechanism is similar to that of other drills already described, and is clearly indicated in the sections. The valve-motion is of the tappet type, the upper arm of the 3-arm tappet *a* engaging with the flat slide valve *b*. This valve controls the main ports *c*, *c* and the exhaust port, which is on one side, opposite the middle of the tappet. A novel feature of the drill is the mode of producing the tappet's

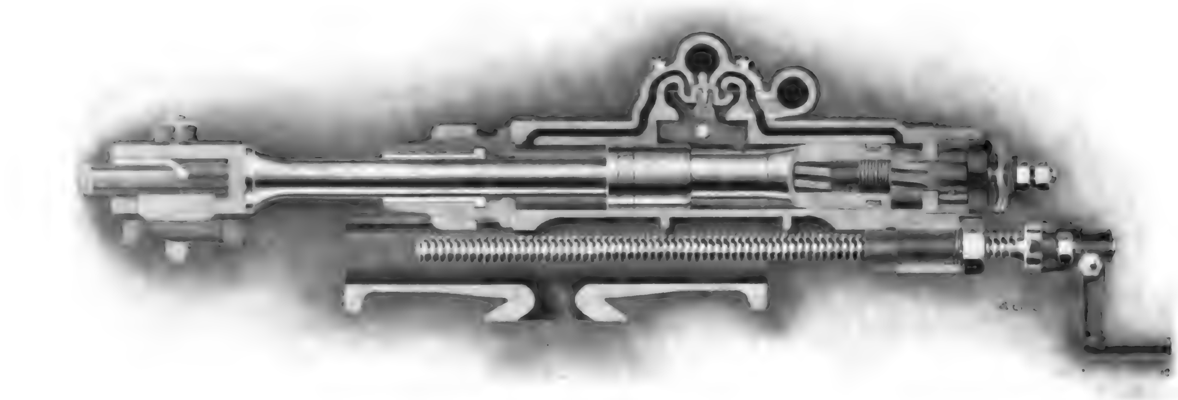


Fig. 139.—Ingersoll-Rand "Arc-Valve" Tappet Drill.

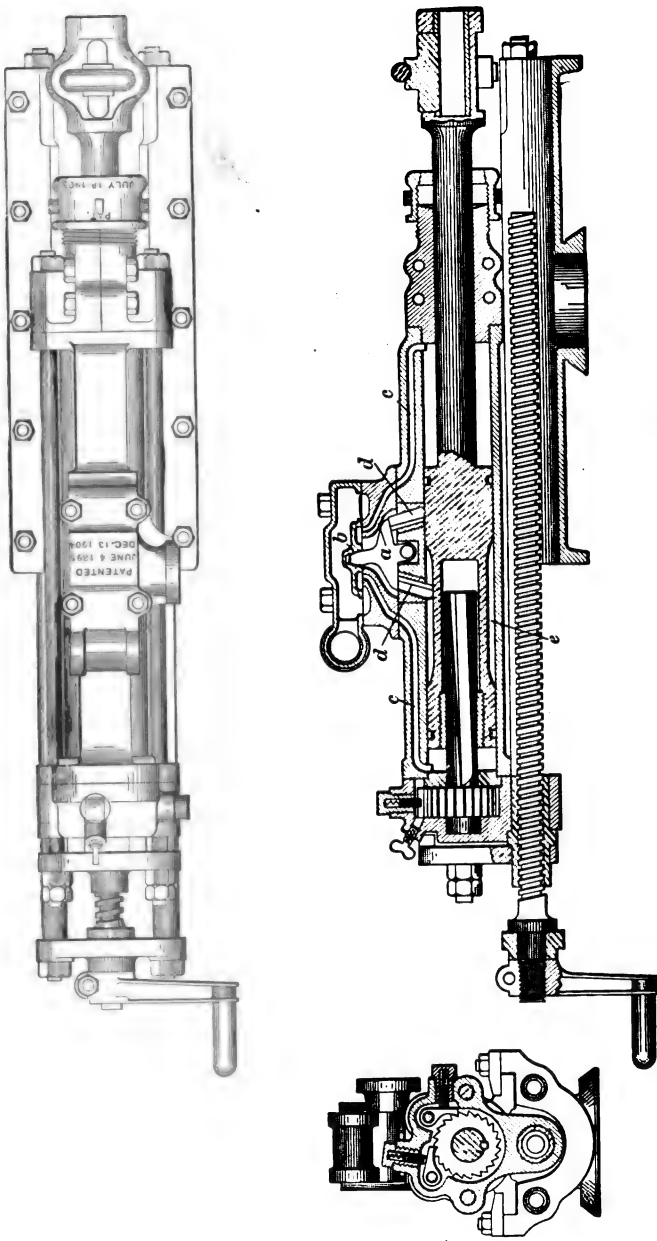
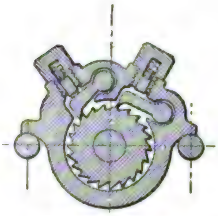
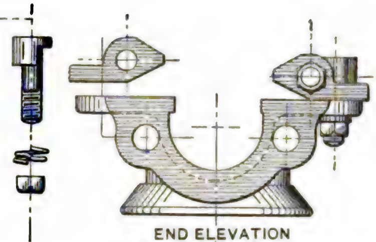


FIG. 140.—Murphy "Little Champion" Drill.

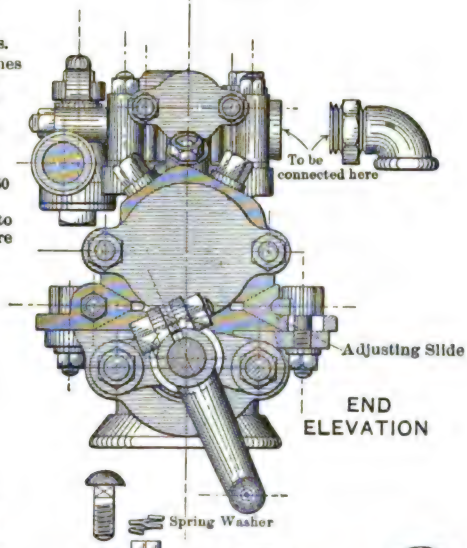


SECTION SHOWING RATCHET WHEEL, PAWLS, PLUNGERS, SPRINGS, AND PLUNGER GUIDES

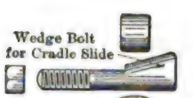


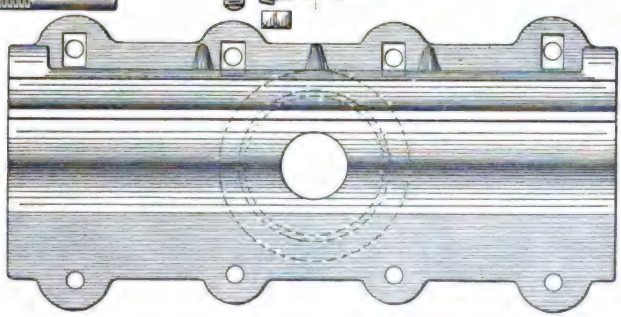
END ELEVATION OF CRADLE

Weight of Drill	=	280 Lbs.
Extreme Length	=	46 Inches
Extreme Height	=	12¼ "
Width of Drill	=	9¾ "
Extreme Width of Drill Including Air Tap	=	10 "
Length of Feed	=	26 "
Strokes per Minute	=	From 450 to 500 According to Air Pressure



END ELEVATION





PLAN OF CRADLE, SHOWING ONE SLIDE AND ALL BOLTS REMOVED

¼-inch.

movements. Its two lower arms do not project into the top of the cylinder bore, to a contact with the piston, as is the case with most drills of this type. Instead, a steel pin *d*, sliding in a bushed hole or seat in the cylinder casting, is set under each tappet arm. As the piston makes its strokes, the curved shoulders of the annular groove *c*, in the piston, alternately strike the rounded ends of the forward and rear tappet pins, *d*, *d*, thus pushing up the pins and causing the tappet to oscillate.

The Murphy drill is made in eight sizes: of $2\frac{1}{4}$, $2\frac{1}{2}$, $2\frac{3}{4}$, 3, $3\frac{1}{8}$, $3\frac{1}{4}$, $3\frac{1}{2}$, and $3\frac{5}{8}$ inch diameter of cylinder, the drill head and shell weighing, unmounted, from 125 to 395 lbs. It is designed for the usual tripod or column mounting.

Climax Imperial Drill. (Fig. 141.) This is a well-known English machine, of the spool- or piston-valve type, made by R. Stephens & Son, Carn Brea, Cornwall. Air enters the valve-chest by the air tap, shown in detail section and also in place on the main elevation. Thence it passes into the annular recess *b*, of the valve (main longitudinal section), which, by its reciprocations, opens communication through the valve-seat ports *c*, alternately with the main cylinder ports *a*, *a*. In the valve are the recesses *d*, shown also in section to the left. These, by the movements of the valve, control the exhaust ports *e*, which connect with the main exhaust on the side of the chest. The throw of the valve is caused by the admission of a little live air through the small grooves *j*, *j*, to the ends of the chest, this air being alternately discharged by the much larger auxiliary ports *f*, *f*. These open into the cylinder through the auxiliary ports *g*, exhausting at each stroke into the annular recess of the piston and thence into one of the square ports *h*, which lead to the main exhaust. The ports *g* are bushed with composition metal rings, shaped at the lower end to fit closely upon the piston.

As shown in the cut, the piston is in position to begin its forward stroke; the valve has been thrown to the right and is admitting air to the rear end of the cylinder. The drills are designed to run at the high speed of from 450 to 500 strokes per minute, according to the air pressure. Other details of construction are shown,

including the cradle or shell, section of the rifle-bar ratchet and its pawls, a new form of chuck and a "dust allayer."

The chuck comprises a heavy wedge and half bushing, with a long bearing surface which grips the bit shank firmly by means of the U-bolt. A tap with a hammer loosens the wedge, permitting rapid changing of bits. When worn, the chuck bushing is set up by a liner, to keep the bit in the axis of the machine.

A specialty of this drill is the "dust allayer" (shown in detail, plan, and elevation), which is attached to the air tap by a nipple and cup, forming a ball-and-socket joint. It is, in effect, an ejector, drawing water from any convenient source, by means of a small quantity of compressed air led from the throttle. By the same air the water is sprayed forward into the mouth of the drill hole.

Stephens & Son build piston valve drills of the

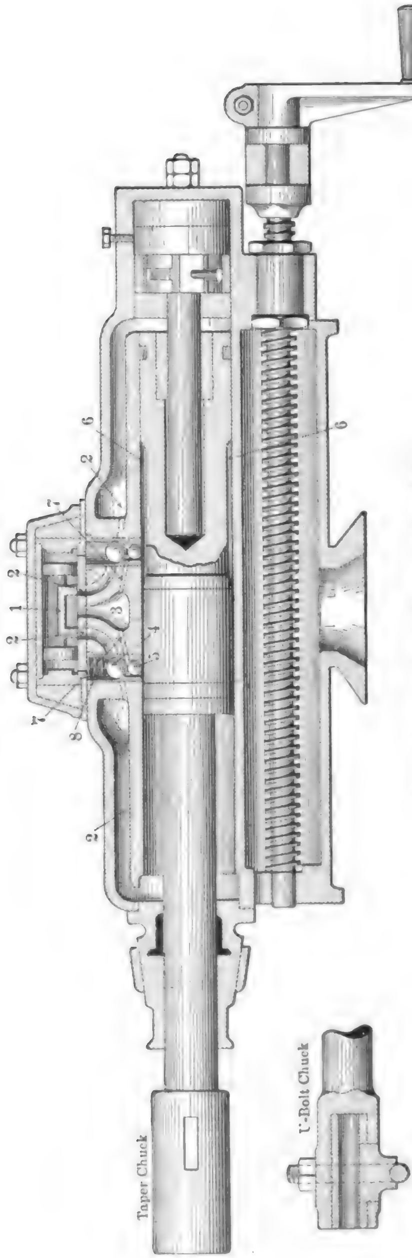


FIG. 142.—Holman Spool-Valve Drill.

following sizes: $1\frac{3}{4}$ in., 2 in., $2\frac{1}{4}$ in., $2\frac{1}{2}$ in., 3 in., $3\frac{1}{4}$ in., and $3\frac{1}{2}$ in.; also tappet valve drills of $2\frac{3}{4}$ in., $3\frac{1}{4}$ in., and $3\frac{1}{2}$ in. diameter of cylinder.

Holman Drill. There are two forms of this English machine, built at Camborne, Cornwall: the air- or spool-valve drill, made in 2, $2\frac{1}{4}$, $2\frac{1}{2}$, $2\frac{3}{4}$, $3\frac{1}{4}$, $3\frac{1}{2}$, and $3\frac{5}{8}$ inch sizes; and the tappet drill, of $2\frac{1}{2}$, $3\frac{1}{4}$, $3\frac{1}{2}$, and $3\frac{5}{8}$ inch diameter of cylinder. The 2 and $2\frac{1}{4}$ inch drills are of light weight, intended chiefly for stoping in thin veins.

Spool-valve Drill. In a few of their details the smaller sizes of this type differ somewhat from the larger, though the general design is substantially the same in all. Fig. 142 illustrates the sizes from $2\frac{1}{4}$ to $2\frac{3}{4}$ inch. The movements of the spool-valve 1, which control main air and exhaust ports of the usual form, are caused as follows: Below each end of the valve-chest, and communicating from chest to cylinder, is a short vertical air port, containing a coned or taper seating. In each of these ports is a pair of steel balls, 4, 5, the former of which controls the auxiliary port 7. Both balls are under the pressure of the spiral spring 8. The seat is so shaped that the lower and smaller ball 5, will project slightly into the cylinder, whenever permitted to do so by the position of the annular recess 6, around the middle part of the piston. Hence, by each stroke of the piston, the lower ball 5 receives a slight upward blow from the inclined shoulder of the recess. This lifts the larger ball 4, also; thereby opening the auxiliary port 7, and placing the corresponding end of the valve-chest in communication with the main exhaust 3. Owing to the pressure of the air occupying the opposite end of the chest, the spool-valve is then reversed, to prepare for the next stroke of the piston. Ball valves are well adapted for this service. They are not liable to breakage, and, as they receive a slight rotary motion from each blow of the piston, the wear tends to be equalized, thus keeping them round and preventing leakage between the balls and their seats.

Tappet-valve Drill (Fig. 143). This differs from most of the American-made drills of the same type in the shape of the tappet and piston. The oscillations of the tappet 10, are caused by the

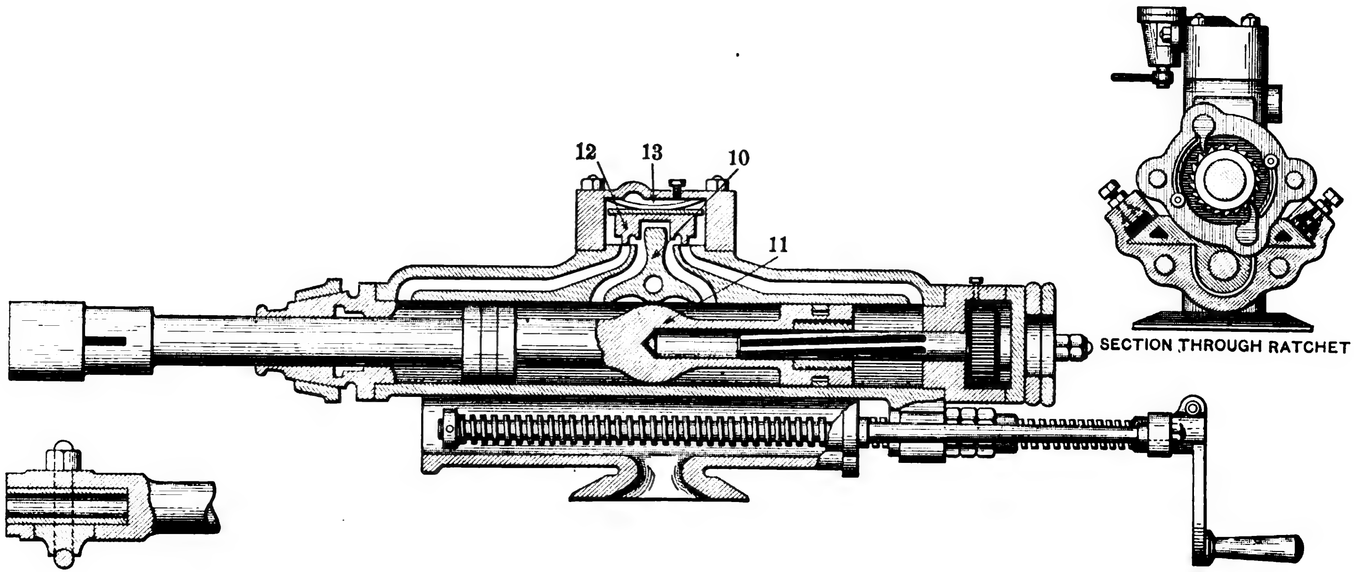


FIG. 143.—Holman Tappet-Valve Drill.

enlarged middle portion, 11, of the piston; the upper arm of the tappet engaging with the D-valve 12. A spring, 13, holds the valve on its seat.

German Drills. A number of good machine drills are made in Germany, most of them designed on the spool- or air-valve principle, and differing chiefly in the smaller details from the American and English machines already described. Amongst them are those of: P. Hoffmann, Eiserfeld; R. Meyer, Mühlheim-Ruhr; Froelich and Klüpfel, Unter-Barmen; Duisburger Maschinenbau, Duisburg; H. Flottmann and Co., Bochum; Freimann and Wolf, Zwickau and the Küzel drill, of R. W. Dinnendahl, Steele. Several of these machines, for example, the Duisburger, use hollow bits and a water jet.

Some repetition would be necessary to describe these drills and details are therefore omitted. A brief description is given below, however, of the "Triumph" drill, built by H. Schwarz and Co. (Ruhrthaler Maschinen-fabrik), Mühlheim-Ruhr. In being valveless and in the mode of distributing and exhausting the air, this machine is interesting in having a strong resemblance to the recent "hammer" drills, and also to the old Darlington drill, for many years well-known in Great Britain.

"Triumph" Drill (Fig. 144). The cut shows a longitudinal section, together with a transverse section through the rotating ratchet and pawls. Air is admitted by the two-way throttle valve *c*, on top of the cylinder, entering thence the annular port *d*. At the beginning of the stroke, as shown in the cut, *d* is in communication with an annular recess *e*, near the forward end of the piston; whence the compressed air passes through *f*, which is one of four longitudinal ports in the body of the piston, to the rear end of the cylinder. The forward stroke then takes place, the air in front of the piston being exhausted through the ports *j*. As the piston advances, the exhaust ports *j* are covered by the solid part of the piston, thus cushioning the end of the stroke. When the annular recess *e*, in the piston, comes opposite the ports *j*, the air exhausts from the back end of the cylinder. At the same time, the annular recess *h*, near the rear end of the piston, comes into

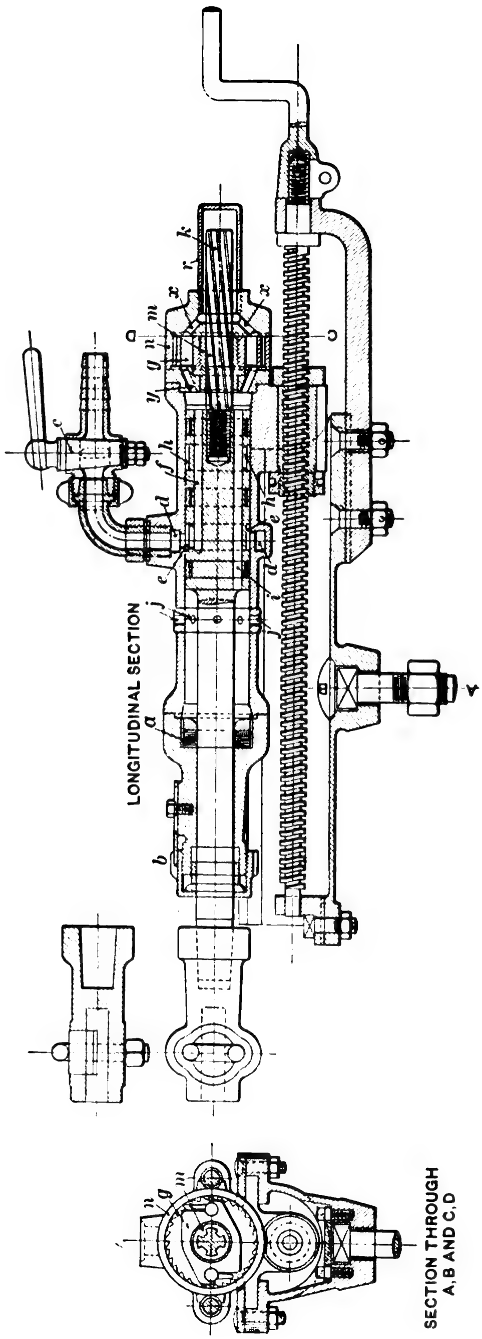


FIG. 144.—"Triumph" Drill.

connection with the air inlet *d*, thus admitting live air through the four longitudinal piston ports, one of which, *i*, is shown in section. These ports conduct the air to the forward end of the cylinder and the stroke is reversed.

The parts are so proportioned that the air acts at full pressure throughout only about one-third of the forward stroke and then expands. Should the piston strike the cylinder head, the shock is absorbed by the spring *a*. In the front head is a stuffing box, the gland of which is held in place by the cap *b*.

Contrary to the usual construction of rotating devices, the rifle-bar *k* is solidly screwed into the rear end of the piston and engages with a correspondingly rifled nut *m*, in the ratchet *g*. The outer end of the rifle-bar reciprocates in the closed tube *r*, which is connected with the cylinder by the small passages *x* and *y*. As a result, live air acts on the entire area of the rear end of the piston, including the area of the rifle-bar. The tube *r*, and connecting passages, serve incidentally for the better distribution of oil on both sides of the ratchet and rifle-nut.

Temple-Ingersoll "Electric-Air" Drill. A discussion of electric rock drills—employing the term in the usual sense—would here be inappropriate. The "Electric-Air" drill is unique in the mode of combining both systems of power transmission and in no way belongs to the class of electric-driven drills, which for years have been brought out from time to time, but which as yet have not given wholly satisfactory results.*

The Temple-Ingersoll machine (Fig. 145) comprises three parts: a drill, and an air pulsator which is driven by an electric motor. Both pulsator and motor are mounted on a small, flat-wheel truck, close to the drill and connected with it by two short lengths of hose. As shown in the cut, the drill differs in many respects from the ordinary rock drills. The cylinder is of larger diameter and the short piston, with packing rings, somewhat resembles the piston of a steam engine. The drill is carried in a supporting and guiding shell, mounted on a column or tripod, and

* It may be pointed out that some resemblance to this machine is traceable in the design of the "Pneumelectric Coal Puncher" (Chapter XXII).

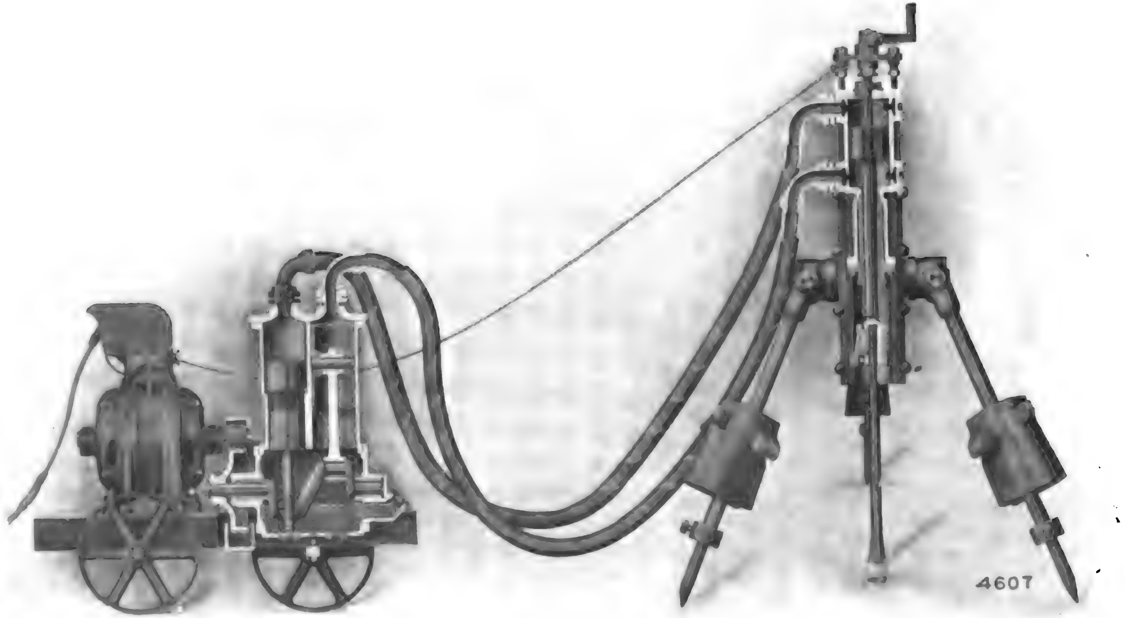


FIG. 145.—Temple-Ingersoll "Electric-Air" Drill.

is provided with a feed screw and rotation device. It is valveless and has no buffers, springs or side rods. The pulsator is in effect a small, vertical, duplex, single-acting compressor, with cranks set at 180°; the crank-shaft being driven through single reduction gearing from the armature shaft of the motor. From the pulsator cylinders one length of hose passes to the back end of the drill cylinder, the other to the forward end. These connections serve as ports for admission and return of the compressed air. There is no exhaust. The air circuit is closed, the same air being used over and over again. Thus the speed of the stroke of the drill depends on the speed of the motor, and this is varied by a controller, operated through a cord by the drill runner. If a direct current motor be used, it is designed for three speeds; or an alternating current motor, single speed, may be employed if desired.

The pulsator runs at a low air pressure, only a small degree of compression being necessary for transmitting the power and acting as a spring between pulsator and drill. Incidentally, the air cushions the drill piston at the ends of the stroke. Leakage of air from joints and past the pulsator pistons is provided for by a compensating valve (not shown in the cut), which is adjusted to maintain a practically constant pressure in the air circuit. When the pressure falls below the limit, the valve opens automatically and admits a little more air. This air is compressed by the differential area between the two parts of the piston in the first cylinder, until the normal working pressure is restored. Lubrication of the pulsator cranks and pistons is effected by the "splash" method, the lower part of the crank-case being partly filled with oil. A portion of the oil is atomized and carried with the air into the drill cylinder.

The "Electric-Air" drill is made in four sizes as shown in Table XXIX.

In working capacity these machines correspond approximately to the 2 in., 2¾ in., 3 in., and 3¼ in. sizes of the Sergeant air drill, described previously. The voltage recommended is 220. For alternating current the standard motors (which are stronger and

TABLE XXIX

No.	Diameter Cylinder, Inches.	Stroke, Inches.	Strokes per Minute.	Weight of Drill Unmounted, Pounds.	Weight of Pulsator and Motor, Pounds.		Approximate H.P. at Pulsator for One Drill.
					D. C.	A. C.	
3–C	$3\frac{5}{8}$	$6\frac{3}{4}$	475	119	525	370	$5\frac{1}{2}$
4–D	$4\frac{3}{4}$	7	415	223	645	545	4
4–E	$4\frac{3}{4}$	7	440	228	883	928	
5–C	$5\frac{5}{8}$	8	400	299	1050	820	3

simpler than those for direct current) are three-phase, 25, 30, 50, and 60 cycle. Direct current motors, wound for 440 or 500 volts, may be used, but these pressures are unnecessary and are dangerous for underground service.

Air Pressure for Machine Drills. The evidence adduced from recorded tests shows conclusively that a low air pressure is uneconomical. Both the force of the blow and the number of strokes per minute fall off, resulting in a marked decrease in the footage of hole drilled. While it is probable that drilling in soft rock does not require so high an air pressure as for hard, it is found on the whole that the best results are obtained by a pressure of from 70 to 80 lbs. Practice of late has tended toward the use of higher pressures, up to 90 lbs. or even more; but, granting that more work in some kinds of rock may be done by employing a heavier pressure than, say, 80 lbs., the life of the drill is shortened and the cost of repairs increased. The customary nearly uncushioned blow, under a heavy air pressure on hard rock, becomes very destructive to the machine, and the bits themselves do not stand so well. They are dulled sooner and are more apt to chip.

The influence of air pressure, as well as the questions relating to air consumption per drill, are further illustrated by a number of tests made several years ago in the South-African gold district.*

* J. B. Carper and others, *Mechan. Engineers Assoc. of the Witwatersrand*, 1904. (Abstract in *Mines and Minerals*, Sept., 1904, p. 64.)

The rock in which the tests were made was red granite, a large block of which was embedded in concrete. A quarry bar was used for mounting the drills. All holes were drilled vertically, with abundance of water. Two receivers were employed, with a combined capacity of 757 cu. ft., the pressure for each run being raised by the compressor to 80 lbs., after which the receiver was shut off. A single machine at a time was operated, the run continuing until the receiver pressure dropped to 70 lbs. The drill was then stopped, and the depth and diameter of hole measured. Similar runs were successively made with pressures from 70 to 60, 60 to 50 lbs., etc. The capacity of the receiver, in terms of cubic feet of free air, was calculated for each individual run and pressure, correction applied for temperature, and the air consumed based on the volume of free air at 70° F. and 24.8 ins. of barometer (equivalent to an altitude of 5,000 ft.).

Eliminating several results of such runs as indicated erratic behavior of the drills, probably due to being in poor condition, a test of 13 drills, 3¼ ins. diameter with 3-in. bits gave the following averages:

TABLE XXX

	Air Pressure, Pounds.				
	80–70	70–60	60–50	50–40	40–35
Linear inches drilled per min.	1.3	1.1	1.0	0.6	0.5
Cu. ft. free air per minute	124.	117.	100.	70.	60.
Cu. ft. free air per linear in. of hole	95.3	106.4	100.	116.4	120.
Ditto per cu. in. of hole	13.3	14.8	13.8	15.0	16.6

Each run occupied about 6 minutes. Some of the average results are not consistent, and the individual figures of course showed still greater variations. These were due to a variety of causes, such as lack of uniformity of the rock, differences in temper and sharpness of bits and, in a measure, the personal equation of the drill-runners, each of whom "was selected by the agent of the maker of the drill." The rather lengthy paper from which these data are taken includes many tables, giving details

of the tests of machines of different makers, and is to be recommended for the thoroughness with which the work was summarized. Among other points, the importance of the question of air pressure is clearly demonstrated.

Consumption of Air. By reason of the irregularity of the work of machine drilling, and the fact that in mining or other rock-excavation work a number of drills are always operated by the same compressor plant, few figures are available as to the actual air consumption of a single machine. Average figures, however, are the only really useful ones. It is customary to base the duty on the consumption of free air per minute, the quantity necessarily depending on the size of the machine, air pressure supplied by the compressor, character of the rock, and the proportion of the total time actually occupied in drilling. It is evident that the compressor capacity for a single machine is greater than the average required for a number of machines. With a large number, the delays to which each is subject, for setting up or shifting, changing bits, stoppages caused by the bit sticking in the hole, etc., make it improbable that all of them will be in simultaneous operation, save in rare instances; hence, the average allowance of air for each may be reduced. Momentary or occasional peaks in the load on the compressor, when an unusual number of drills happen to be working simultaneously, may be disregarded; or at least need not be provided for by increasing the compressor capacity.

Rock-drills of different makers, even when of the same diameter of cylinder, vary in their consumption of air and reliable figures are not easily obtained. Table XXXI, showing the volume of free air per minute required for one drill, is based on a comparison of the statements of several manufacturers, checked by a few recorded tests. It may be taken to represent, within reasonable limits of error, the results of actual practice for machines in good order. No allowance is made for the preventable loss of air in leaky pipes, nor for frictional loss of pressure in transmission (see Chapter XVI).

TABLE XXXI

CUBIC FEET OF FREE AIR PER MINUTE CONSUMED BY ONE DRILL AT SEA-LEVEL

Gauge Pressure.	DIAMETERS OF DRILL CYLINDER IN INCHES.											
	2	2¼	2½	2¾	3	3⅛	3 3/16	3¼	3½	3⅝	4¾	5
60	58	63	70	82	90	97	100	105	114	118	135	155
70	62	72	80	92	104	112	115	118	130	135	152	174
80	70	80	88	103	115	125	130	135	142	153	173	205
90	78	87	95	115	128	137	141	148	165	173	194	222
100	85	96	108	126	140	151	155	161	176	184	210	250

When a number of drills are operated by the same plant, the compressor capacity for furnishing the total average quantity of free air required per minute, at sea-level, may be found approximately by the following table of multipliers:

TABLE XXXII*

Number of drills...	1	2	3	4	5	6	7	8	9	10
Multiplier.........	1	1.8	2.7	3.4	4.1	4.8	5.45	6.1	6.7	7.3
Number of drills...	11	12	15	20	25	30	35	40	50	60
Multiplier.........	7.8	8.4	10.3	12.8	15.1	17.3	19.7	22.0	26.5	30.5

The required capacity of the compressor is found by multiplying the cubic feet of free air per minute consumed by a single drill (as given in Table XXXI), by the multiplier corresponding to the number of drills operated (Table XXXII).

It will be understood from what precedes that the figures in the tables cannot be taken as exactly applicable to all cases. Several other modifying factors may here be summarized:

(1) **The Kind of Work.** The time required to set up the drill depends greatly on the shape of the working, whether a tunnel or drift, a shaft, stope, or open cut. If the floor and roof, or the side walls, of a mine opening are irregular or loose, much time may be

* Based on comparison of several tables given by manufacturers.

lost in shifting the machine and setting it up, according as it is mounted on column or tripod.

(2) **Character of the Rock.** This also influences the consumption of air. In hard rock the rate of advance in drilling is slower than in soft, so that the machine makes longer continuous runs. Less total time is occupied in shifting and setting up for drilling the successive holes of a round, and the consumption of air per unit of time is therefore greater. Though this increase is partly offset by the fact that the bits are more quickly dulled in hard rock and must be changed at shorter intervals; still, in very hard ground the machines may be kept running with but few and short intermissions. In soft rock, on the other hand, though the actual speed of drilling is greater, there are apt to be more frequent delays due to rifling of the hole and sticking or "fitchuring" of the bit. On the whole, for hard rock it is advisable to provide a greater compressor capacity than is given in the tables. The compressor will then be able to run at a slower speed, thus avoiding excessive heating in cylinder and receiver. In general, the time actually occupied in drilling will vary for each machine from, say, 4 to 6 hours out of an 8-hour shift.

(3) **Physical Condition of the Drill.** The importance of this matter may be overlooked. The figures given are for new machines, or those in thoroughly good order. More air is consumed by old drills, whose valves and pistons are so worn that they do not fit closely. Even in the case of drills in fair average condition, this is clearly shown by the fact that the exhaust, instead of being short and sharp, is nearly continuous. A large allowance must be made for old machines.

If definite values could be assigned to these different items, estimates of air consumed per drill could be made in conformity with any given set of conditions. To do this is manifestly impossible, but a few general data relative to averages for an entire shift's work have been put on record by Messrs. J. E. Bell and L. L. Summers, as the result of a series of experiments (*Mining and Metallurgy*, Feb. 1st, 1901). For a 3-in. drill, the volume of free air required per shift of 8 hours is as follows, the gauge pressure being 100 lbs.:

TABLE XXXIII

Elevation.	Cubic Feet of Free Air.	
	Per Shift of 8 Hours.	Per Minute.
Sea-level	25,000 to 42,000	52.1 to 87.5
5,000 ft.	30,000 " 49,000	62.5 " 102.0
10,000 ft.	35,000 " 60,000	73.0 " 125.0

These figures include all deductions, for whatever cause, covering delays and stoppages as well as the actual drilling time.

Taking the various allowances into account, and applying them to Tables XXXI and XXXII, the following results, obtained in an elaborate test made at the Rose Deep Mine, Johannesburg, South Africa,* will be found in fairly close agreement with what precedes. The average number of drills (Ingersoll-Sergeant), of several different sizes, kept in operation during the 6-hour test, was calculated to be equivalent to 30.9 drills, $3\frac{1}{4}$ in. diameter of cylinder. The average duty per drill was 4 ft. $5\frac{11}{16}$ ins. of hole per hour (diameter of hole not stated). Average air pressure, 69.83 lbs. Free air used per drill per minute, 81.08 cu. ft. It is fair to assume that most of these drills were more or less worn, or at least not in perfect condition. According to the tables, the average free-air consumption for 30.9 drills should have been about 68 cu. ft. per minute, or about 15 per cent. less than that shown by the test. This difference is accounted for in part by the altitude above sea-level. It may be added that the horse-power per drill developed in the steam cylinders of the compressor was 12.72. But as the work done during the 6-hour test was approximately equal to that usually accomplished in 8 hours of regular work, the actual horse-power per drill under normal conditions in this mine may be taken as $12.72 \times \frac{6}{8} = 9.54$. The air piping in this case was known to be remarkably free from leaks.

Another test run, on 75 drills, $3\frac{1}{8}$ in. diameter, was made

* L. I. Seymour, *South African Association of Engineers*, 1898.

about 5 years ago at the Champion Iron Mine, Mich.* At 78 lbs. normal gauge pressure the average air consumption for the day shift, throughout a period of 1 month, was 67.1 cu. ft. of free air per minute. The air pressure usually dropped considerably, however, when work was in active progress. According to the tables, 75 drills should have used an average of about 58.5 cu. ft. of free air per minute, or 13 per cent. less than shown by the test.

Efficiency of Machine Drills. Though it is a well-known fact that compressed-air drills are uneconomical machines in consumption of power, it is difficult to reach definite conclusions as to their efficiency. The actual useful work—employing this term in its ordinary mechanical sense—done by a machine drill in making a hole of given depth and diameter in a rock of given hardness, toughness, and general physical character cannot be determined absolutely. All that is really known is that the drill requires a certain volume of air per minute, which has been furnished by the expenditure of a certain average indicated horse-power at the compressor. Comparative figures of work done, in terms of speed of drilling in a given rock and per cubic foot of free air consumed, are often published and are useful as far as they go. In fact, this is the only practical basis for estimating their efficiency. But, even in this sense, the propriety of accepting the results obtained, as accurately representing the value and efficiency of machine drills as compared with various forms of air or steam engines, may well be questioned.

In their operation rock-drills differ greatly from other compressed-air machines, because the personal element of the skill and experience of the drill-runner exerts so important an influence upon the amount of work accomplished, and because the rate of drilling is so greatly modified by the physical and mineralogical character of the rock, together with the purely adventitious occurrence of cracks, slips, and fissures. A skilful drill-runner will inevitably do more work per shift, under the same conditions, than an inexperienced man, and he will make a faster rate of advance

* *Engineering and Mining Journal*, May 18th, 1905, p. 937.

in a rock with which he is specially familiar than if called on to operate a machine in rock that is new to him.

Therefore, though mechanical efficiency, pure and simple, is the basis upon which machines in general are compared, in the case of compressed-air drills it is not the only consideration, nor is it the most important. Their efficiency of operation is subordinate to the attributes of strength, simplicity of construction, portability, durability, ease and readiness with which repairs may be made and capacity for work in terms of depth of hole drilled per unit of time. They must be capable of withstanding hard and often unintelligent usage. The strong point of compressed-air drills is their ready applicability in the special and peculiar field of work for which they are designed. In possessing a cylinder, piston, and valve, the drill roughly resembles a steam engine, but there the likeness ceases. Severe shock and vibration are essential accompaniments of its work. No fly-wheel is admissible, or other means of storing up and equalizing the power, and the service demanded from the rock-drill is therefore totally different from that performed by ordinary engines.

The low theoretical efficiency of the compressed-air drill is due mainly to the fact that air is admitted to the cylinder practically throughout the full stroke. As a consequence, the valve motion bears a strong resemblance to that of many of the single-cylinder, direct-acting pumps. Expansive use of the air to any extent is neither advisable nor practicable, both because of the undesirability of introducing complexity of mechanism in machines subjected necessarily to rough usage and because of the difficulty of adapting cut-off gear to the variable length of stroke required. Owing to the nature of its work, the drill cannot be kept always at full stroke. While in operation it is often necessary to feed the machine so far forward that the actual length of stroke may be little more than one inch, and the valve motion must still be capable of reversing promptly. A sharp, quick reversal of the stroke is essential. The useful work is done on the forward stroke, in striking the blow. If the valve be thrown

too soon, the stroke of the piston will be shortened; if too late, the piston may strike the cylinder head. For these reasons, it is impracticable with machine drills to attain the economy resulting in other air motors from using the air expansively. Incidentally, the use of air at full stroke is of some advantage, because, in exhausting at high pressure, the exhaust air issues from the port at a high velocity, and its force, combined with the development of some heat from friction, in a measure prevents troublesome accumulation of ice, in case the air is moist. The freezing, if any, is at least confined to the exterior portion of the exhaust port, whence it is easily removed.

In dry, dusty mines it is generally found that the tappet valve gives the better service. When a compressed-air drill is not in use, and disconnected from the air hose, dust and grit are likely to enter through the ports, passing thence into the valve chest and cylinder on resuming drilling. The wear and consequent looseness in the fit of the moving parts thus caused is apt to have a more unfavorable effect on the operation of the spool than the tappet valve. Leakage of air past the valve or piston prevents proper action of the auxiliary ports, not only producing irregularity in reversal and shortening of the stroke, but diminishing the drill's efficiency. It is true that the tappet valve involves the use of one extra part and, in case of the three-arm tappet, breakage is not infrequent. But while the spool valve is strong and reliable, experience indicates that in dusty mines at least the maintenance cost of the spool-valve drill is higher than that of the tappet drill.

The maximum force of blow is attained by drills which take air throughout the full forward stroke, *i.e.*, without cut-off, and the best drills are designed to work in this way. On the forward stroke the valve is not reversed until the blow is delivered, the exhaust being free, with but little back-pressure on the piston. Cushioning was formerly made a feature of some rock-drills, with the idea of reducing shock, but it is now recognized that the efficiency is increased by delivering an uncushioned blow. It is possible for a drill so designed to strike too heavy a blow in very

hard rock, but the remedy then is to feed the drill-head down, so as to work with a shorter stroke.

On the back stroke cushioning is desirable, to ease the reversal and prevent injury by the piston striking the rear cylinder head. The back-stroke cushion is produced by cutting off the exhaust before the end of the stroke. Only enough power needs to be developed on this stroke to overcome the resistance due to the weight of the moving parts, and the frequent tendency for the bit to stick fast in the hole.

In ordinary machine drills, the piston speed should not be too great—say, not much over 350 to 375 strokes per minute. The relative speeds of stroke do not constitute a proper basis for the comparison of efficiencies. To give effect to the blow, the weight of the moving parts must be relatively great, and a very high speed would be attended by excessive wear and breakage. These conclusions do not apply, however, to the numerous small air hammer drills which have come into favor in the past few years. (See Chap. XXI.) The hammer drill strikes a light blow, some of them at the rate of 2,000 to 3,000 or more strokes per minute. Thus the weight of the moving parts is small, and the inertia moderate.

Drill Repairs. There are a number of good machine drills on the market, whose relative merits it is difficult or impossible to determine. In choosing a drill the question of repairs is of great importance. But very little useful information concerning this point is available; and in fact such data could only be obtained by operating under the same conditions, and for a considerable period of time, drills of several different kinds. Such opportunities rarely exist. It is not usual, nor generally advisable, to use different makes or more than two sizes of drill in the same mine or surface excavation, since this practice involves the necessity of keeping duplicate spare parts for each.

The item of repairs depends largely upon the experience and character of the drill runner. A careful man will treat his machine with intelligent consideration. He will set it up properly, to reduce the risk of getting out of alignment as the hole is deepened;

and, if the bit should stick ("fitcher"), will keep his temper and refrain from striking unnecessarily heavy blows on the drill head or chuck. A fitchered bit may often be loosened by slacking the clamp bolts, thus allowing the machine slightly to shift its position. The serious abuse to which machine drills are frequently subjected may be reduced by efficient supervision on the part of foremen and shift-bosses.*

It is desirable, in every piece of machinery, that there shall be as few moving parts as possible. But, in a machine performing the severe work of a rock-drill, and often operated by incompetent men, there is another consideration. Even when run with care, wear is rapid and breakages are frequent. The maker of a machine drill, therefore, has the problem before him of producing a drill consisting of as few parts as practicable; designing it, at the same time, so that those parts which experience shows to be specially liable to wear, derangement or breakage, may be replaced readily, cheaply and without the necessity of discarding perhaps a much larger piece, with which the broken part may be connected. A properly equipped repair shop is a matter of importance for lengthening the life of machine drills. New cylinders may be bored out to fit worn pistons, and new pistons fitted to old cylinders. Drill runners should not be encouraged to tinker their machines underground. If repairs or adjustment be necessary, the drill should be sent at once to the shop.

Records of Work. If it were possible to secure approximately complete records, tabulations would best show the comparative speeds of drilling in different rocks and ores. But the local conditions are obviously so variable that no summary comparison can justly be made. In the following pages I have attempted to elucidate the subject by citing a number of cases. A few of these are the results of observations by the author; most of them have been obtained from engineers in the field and from the detailed notes taken by mining students, who have studied at different mines under the direction of the author. Unless otherwise stated,

* For a good article on the operation of machine drills see *Mining and Scientific Press*, 1905, November 4, p. 308; November 11, p. 329.

the "total time" in each case includes all ordinary delays in drilling, for changing bits, etc., except the time occupied in setting up the machine. This usually takes from 15 to 30 minutes.

San Pedro Copper Mine, N. M. A 5 ft. x 7 ft. cross-cut, in very hard quartzite and limestone; 10-hour shift; one Ingersoll-Sergeant 3-inch drill; average air pressure, 60 lbs.

Heading advanced in one month	38.3 ft.
Number of drill-hours	357
Total number of holes drilled	248
Total number of feet of hole	770
Average depth of hole	37 in.
Average number of feet of hole drilled per hour	2.16

A 5 ft. x 7 ft. drift in the same mine, in quartzite and vein matter, with the same drill and air pressure, was advanced 46.6 ft. in one month:

Number of drill-hours	207
Total number of holes drilled	113
Total number of feet of hole	430
Average depth of hole	45 in.
Average feet of hole per hour	2.08

In the above records all delays for breakage, setting up the machine, and changing bits, are included.

TABLE XXXIV.

Hole.	Depth, Feet.	Total Time, Including Delays. Minutes.	Net Drilling Time. Minutes.	Feet per Minute.	
				Total Time.	Net Time.
1	4.7	50	38	.094	.124
2	5.7	63	56	.090	.102
3	6.4	70	56	.091	.114
4	4.9	48	37	.102	.132
5	5.3	44	39	.120	.136
6	6.0	57	50	.105	.120
7	6.0	90	72	.066	.083
8	5.0	56	44	.089	.113
9	5.0	45	37	.111	.135
10	4.5	47	43	.096	.105
11	6.3	30	27	.210	.231
12	5.0	43	31	.116	.161
Averages	5.4	53.6	44	.107	.129
Average inches per minute				1.29	1.55

Snowstorm Mine, Idaho. A drift in hard quartzite; 8-hr. shift; Ingersoll-Rand 3⅛-inch drill; air pressure, 80 lbs. (Table XXXIV).

Two Tunnels for U. S. Reclamation Service, North Yakima, Wash. Tunnel section, 7 ft. x 7 ft. 3 in., in fissured, hard but blocky basalt; one Wood 3-inch drill; 8-hour shift.

Average advance per month	143.5 ft.
Number of feet of hole per shift	29
Average depth of hole drilled per hour	3.62 ft.
Average cost of drill repairs per shift	58 cents.

Michigan Copper Mine, Rockland, Mich. Stoping in fairly hard amygdaloidal and brecciated vein rock; Rand 3⅛-inch drill; 60–65 lbs. air pressure; 8-hour shift.

Depth of holes	6 to 8 ft.
Total depth of hole drilled in 1 month (26 days)	554 ft.
Average depth drilled per shift	37 ft.
Average depth drilled per hour	4.63 ft.

At the same mine the following observations were made on the drilling of individual holes, in drifting and stoping:

TABLE XXXV.

Hole.	Depth. Inches.	Total Time, Including Delays. Minutes.	Net Drilling Time. Minutes.	AVERAGE INCHES PER MINUTE.	
				Total Time.	Net Time.
(Drifting.)					
1	32	25	22.5	1.28	1.4
2	19	14.5	6	1.30	3.1
3	53	37	14	1.40	3.7
4	52	48.5	22	1.07	2.5
Averages	36	31.2	16.1	1.26	2.7
(Stoping.)					
1	74	62	52.5	1.2	1.4
2	70	36.5	26	1.9	2.7
3	76	..	61.75	..	1.2
4	66	..	29	..	2.3
5	76	..	36.5	..	2.1
6	73	51.5	33.5	1.4	2.3
7	81	35	23	2.3	3.4
8	64	..	19	..	3.5
9	68	53	20	1.3	1.7
Averages	72	47.6	33.5	1.6	2.3

A Lead Mine, near Flat River, Mo. Stoping and drifting in rather hard, tough, bedded limestone, interstratified with bands of shale; formation has numerous cleavage planes, generally parallel. Sullivan drills are used; U. S. 2¼-inch for stoping and U. C. 2¾-inch for drifting; air pressure at drills, 75 to 85 lbs. Average depth of hole, 7 feet. Average footage drilled per 8-hour shift, 39 feet.

The Waugh drifting, stoping, and sinking drills, Nos. 8 and 8 D, with hollow steel, were tried at this mine, but were found to be unsuited to the conditions. They made, respectively, 36.1 ft., 43.7 ft., and 22.3 ft. per 8-hour shift.

Federal Lead Co., Flat River, Mo. Breast-stoping in fairly hard and tough limestone, with 2¼-inch Sullivan drills. Air pressure at drills, 75 lbs. Average depth of holes, 6 ft. 8½ in. The drills were new; work done by contractors. In direction, nearly all of the 25 holes ranged from about 45° above, to 30° below, the horizontal (three were nearly vertical).

TABLE XXXVI.

Number of Run.	Number of Holes.	Total Depth Drilled.		Total Time.		Net Drilling Time.		Per cent. of Shift Actually Drilling.	Drilled per Minute, Net Time, Inches.
		Feet.	Inches.	Hrs.	Mins.	Hrs.	Mins.		
1	4	19	10	3	36	1	56	48.3	2.05
2	8	57	0	7	18	4	06	51.3	2.77
3	6	42	10	6	24	3	50	47.9	2.23
4	7	48	8	7	47	3	48	47.5	2.56
Av. depth of hole,		6	8½	Average speed of all holes,					2.46

Mount Hope Iron Mine, Wharton, N. J. Underhand stoping in solid, magnetic ore; Sullivan 3-inch drill; air pressure, 75 to 85 lbs.

1st hole, 94 ins. deep, in 68 mins. = 1.38 in. per minute.

2d hole, 66 ins. deep, in 53 mins. = 1.24 in. per minute.

Time includes changing bits and other incidental delays, but not setting up the machine.

Detroit Copper Co.'s Mines, Morenci, Ariz. Stoping with

Ingersoll-Rand "Sergeant C-24," 2¾-inch drill, in ore of varying hardness and mineralogical character, as stated below; average air pressure, 75 lbs.

TABLE XXXVII.

Number of Holes.	Aggregate Depth, Feet.	Average Depth, Feet.	Total Time, Hours.	Average Speed of Drilling. Per Hour, Feet.	Average Speed of Drilling. Per Minute, Inches.	Character of Ore.
80	367	4.59	54	6.80	1.36	Soft porphyry.
91	417	4.58	59.5	7.01	1.40	Rather soft quartzite.
74	238	3.21	88	2.70	0.54	Very hard, tough, quartzite.

The above figures represent six weeks' work.

El Oro Mining and Railway Co. } *Mexico.* Approximate drill-
The Mexico Mines of El Oro. }
ing speeds, in stoping in very hard, tough quartz, and for development work (drifting, cross-cutting, and raising), in quartz, andesite, and hard, black shale or slate. Drills used: Ingersoll-Rand "Sergeant C-24," 2¾-inch and "Sergeant A-86," 2¼-inch; air pressure, from 70 to 75 lbs.

2¾-inch drill: average speed of drilling, including changing bits but not shifting and setting up, 1.5 in. per minute; average footage per 8-hour shift, 35 ft.

2¼-inch drill: average speed (as above), 1.2 in. per minute; average footage per shift, 30 ft.

Wabana Iron Mines, Nova Scotia. Stoping in red hematite (hard-ore type); Sullivan 3-inch, spool-valve drill; 70–75 lbs. air pressure; 10-hour shift.

Depth of holes	6 to 8 ft.	
	One Month	One Month
Number of drill shifts	340	372
Average feet of hole drilled per shift	58.6	69
Average feet of hole drilled per hour	5.8	6.9
Average cost of drill repairs per shift	45.4 cents.	

Pennsylvania Copper Mine, Butte, Mont. Stoping; ore chiefly a compact granitic and quartzose gangue, with streaks and bunches of chalcocite, moderately hard; Rand 2¾-inch drill; 8-hour shift.

TABLE XXXVIII.

Hole.	Depth, Inches.	Total Time. Minutes.	Net Drilling Time, Minutes.	Average Inches per Minute.	
				Total Time.	Net Time.
1	62	30.5	22	2.00	2.80
2	66	46.5	27	1.42	2.44
3	66	59	38.5	1.12	1.66
4	64	....	32	....	2.10
Averages	64.5	45.3	30	1.51	2.25

Highland Boy Mine, Bingham, Utah. In a 10 ft. by 10 ft. drift, fairly hard, compact limestone, with Rand 3¼-inch drill, the average drilling speed per hour, for several observations, was 2.43 feet of hole, including all delays except setting up. One round of 13 to 14 holes, 4 ft. deep, are drilled and blasted in this drift in one 8-hour shift. For stoping in pyritic ore, with a 3⅛-inch Rand drill, a round of four 4-ft. holes are drilled in 3¼ hours, or at the rate of 5.1 ft. per hour.

Vekol Gold Mine, Pinal Co., Ariz. Results of a drilling contest in a 4½ by 6 ft. drift, in solid blue limestone; Sullivan 2¾-inch spool-valve drills; 105 lbs. air pressure; 8-hour shifts.

TABLE XXXIX.

Shift.	Time for Setting Up, Minutes.	Number of Holes.	Feet Drilled.	Average Depth of Hole, Feet.	Total Working Time.		Average Speed of Drilling, Feet per Hour.
					Hours.	Mins.	
1	26	11	94	8.5	7	54	11.9
2	16	11	95	8.6	6	25	15.2
3	19	12	102	8.5	7	15	14
4	21	10	85	8.5	7	10	11.8
5	17	12	92	7.7	7	02	13.1
Totals	99	56	468	...	35	46	....
Averages	19.8	11.2	93.6	8.36	7	09	13.2

The drift was advanced at the average rate of 6 ft. 2 in. per shift. This extremely rapid work is interesting in showing what

can be done under the stimulus of the spirit of rivalry produced by a "drilling contest."

Wolverine Copper Mine, Michigan. The following results of drilling in the characteristic amygdaloidal vein matter of the Keweenaw copper district are abstracted from a paper by W. R. Crane (*Engineering and Mining Journal*, Sept. 8, 1906, pp. 438–9). Rand drills were used, of 3 and 3⅛-inch diameter of cylinder.

Table XL.

Kind of Work.	Average Depth, Feet.	Total Average Drilling, Time per Hole.		Net Average Drilling, Time per Hole.		Average Inches per Minute.	
		Min.	Sec.	Min.	Sec.	Total Time.	Net Time.
Drifting, 10 holes	5.76	52	23	36	44	1.31	1.88
Drift stoping, 9 holes	5.60	41	47	24	04	1.60	2.80
Cutting out stope, 6 holes	7.50	48	40	29	54	1.85	3.01
Raise stoping, 9 holes	6.50	52	12	41	00	1.50	1.90
Averages	....	....	....	....	....	1.56	2.40

Table XLI.—(Drifting).

Hole.	Depth, Feet.	Total Time, Minutes.	Net Time, Minutes.	Average Speed, Inches per Minute.	
				Total Time.	Net Time.
1	5.5	31	26	2.13	2.54
2	5.5	32	27	2.06	2.44
3	5.5	37	30	1.78	2.20
4	5.5	38	31	1.73	2.13
5*	6.0	148	100	0.50	0.72
6	6.0	50	40	1.44	1.80
Averages	5.66	56	42	1.61	1.97

* Hole No. 5 was seriously delayed by "fitchering" of the bits. Omitting it, the average speed of the other five holes is 1.83 in. per minute for total time and 2.22 in. per minute for net time.

TABLE XLII.—(STOPING).

Hole.	Depth, Feet.	Total Time, Minutes.	Net Time, Minutes.	Average Speed, Inches per Minute.	
				Total Time.	Net Time.
1	7.5	46	38	2.00	2.37
2	7.5	54	45	1.66	2.00
3	6.75	64	57	1.26	1.42
4*	7	115	87	0.73	0.96
5	8	45	38	2.13	2.53
Averages	7.35	65	53	1.56	1.86

* Hole fitchered. Omitting it, the average speeds are respectively 1.76 and 2.08 in. per minute.

Portland Gold Mine. Cripple Creek, Colo. Stoping in very hard phonolite-breccia, with Ingersoll 2¼-inch drill; conditions difficult.

TABLE XLIII.

Hole.	Depth, Inches.	Total Time, Minutes.	Net Time, Minutes.	Average Speed, Inches per Minute.	
				Total Time.	Net Time.
1	18	26	22	0.69	0.82
2	38	28	24.25	1.36	1.57
3	25	46	42	0.54	0.60
4	24	33	31	0.73	0.80
5	30	45	..	0.66	...
6	33	23	..	.44	....
7	18	26	..	.69	....
8	38	36	..	1.05	....
Averages	28	33	(4 holes) 29.8	0.89	(4 holes) 0.95

The above record is below the average for stoping in this mine. Where the ground is more favorable, eight 3-ft. holes are usually a fair day's work.

In a drift of the *Cresson Mine*, also at Cripple Creek, observations of two holes, 53 inches deep, drilled by a 2¼-inch Ingersoll

drill, showed the high speeds of: 2.2 and 3.0 in. per min. for total time; 3.1 and 3.65 in. per min. for net drilling time.

Conclusions. The average speed of drilling shown by the sixteen examples given in the preceding pages is 6.2 feet per hour. In general it may be said that the duty of a standard 3-inch machine drill, in rock or ore of average hardness, ranges from 40 to 50 feet per 8-hour shift, including all ordinary delays for setting up and changing bits. For very hard, tough ground, the speed is often much lower; while considerable more than 50 feet per shift may be made when the conditions are favorable, and also in drilling deep holes, for which fewer set-ups are required. The cost per foot of hole is obviously extremely variable, ranging from say 8 cents in easy ground, and where wages are low, up to 25 cents, when the conditions are adverse.

For moderately soft ground, not requiring holes of large diameter to contain the necessary quantity of powder, the smaller sizes of machine drill—from 2 in. to 2½ in.—are usually preferable. Their first cost, as well as air consumption, is less than for large drills, and they may usually be operated by one man. These small machines are specially useful for stoping in rather thin veins. But, for hard ground, and as a rule in shaft sinking, tunneling, cross-cutting and similar work, the 2¾ in., 3 in., and 3⅛ in. sizes are best. For deep holes in large surface excavations, still heavier drills are often necessary—up to 3½ in., or even larger.

CHAPTER XXI

COMPRESSED AIR HAMMER DRILLS

The principles of the hammer drill were first applied in pneumatic riveting hammers, and tools for chipping, rough chiseling and miscellaneous machine shop work. Their earliest employment in mines was for cutting hitches for timbers, block-holing, and other small work, where no great depth of hole is required.* In recent years they have rapidly grown in favor, as the result of the introduction of improved designs, and are now competing with the larger and heavier reciprocating machines for all kinds of comparatively shallow drilling, like quarry work and stoping in thin veins; also for sinking shafts and winzes where it is desirable to use small and light machines.

All the different makes of hammer drill are built in several sizes and patterns. The smaller machines weigh only 18 lbs., and are held in the hands of the operator, a D-shaped handle being provided for convenience (Fig. 148). These light drills are designed for block-holing and sinking, and in general for holes directed below the horizontal. Many of the larger sizes are designed with an air-feed standard, for overhead work (Fig. 151), and may or may not be mounted on a light column. These, too, are easily carried from point to point and set up and run by one man. Still others, like the Kimber, of South Africa, are, in diameter of cylinder and weight, about equal to the small sizes of reciprocating drills. Lastly, the Leyner hammer drill, which may be said to form a class by itself, is built of the same sizes as the average standard machines of the reciprocating pattern.

* The writer believes that the Franke hammer drill, brought out in Germany in 1891 or 1892, was the first hammer drill used in mining. (See *Zeitschr. für das Berg-Hütten-und Salinenwesen*, Vol. 41, p. 110.) It weighs 16 lbs. and strikes several thousand blows per minute.

General Construction. In the hammer drill, the bit does not reciprocate. The shank of the bit projects into the forward end of the cylinder and is struck a rapid succession of blows by the piston, which acts as a hammer. When in operation the cutting edges of the bit are in constant contact with the bottom of the hole, except during the slight rebound caused by each blow of the hammer. Many of these machines are valveless, the functions of the valve being performed by the reciprocations of the hammer or piston. Others, like the Leyner, Sullivan, Climax, and one of the types of the Ingersoll-Rand, are provided with spool-valves. With the exception of the Leyner hammer drill and one or two others, no attempt is made to introduce automatic rotation of the bit. Rotation in all the smaller machines is effected by hand, the operator turning the whole machine back and forth on its axis, by means of the handle. The bit shank is made octagonal, generally fitting loosely in the chuck socket, which is of the same shape. To keep the hole round and reduce the chances of rifling, the bit is commonly of the star shape, with six (sometimes eight) radial cutting edges. This construction brings the cutting edges so close together that even if several successive blows are made in the same position of the bit, the ridges of rock between the edges are broken away.

As the bit does not reciprocate, it is evident that, for holes directed downward and more than a few inches deep, some automatic means must be provided for removing the drill dust or sludge and keeping the bottom of the hole clean, otherwise much of the useful effect of the blows of the hammer would be lost. To accomplish this, a hollow bit is generally used, a small hole being bored longitudinally through its center. By injecting a jet of water, the drillings are displaced and the bit is kept cool. The same result is attained by a jet of compressed air, which produces a low temperature on expanding. As the speed of stroke of hammer drills is great, the cooling of the bit in dry holes is important. The air jet is obtained by exhausting through the bit, at each stroke, part or all of the air from the cylinder. When using water, it may be led to the chuck through a special passage in the

cylinder; or a small tube is inserted longitudinally through the center of the piston, to the inner end of the bit. The water is supplied by gravity, or under artificial pressure (as for the Leyner drill).

The small hand hammer drills are fed simply by keeping the bit pressed firmly against the bottom of the hole. In these, much of the shock and vibration are absorbed by a handle of rubber hose. Some of the larger machines of all the principal makes are provided with an automatic air-feed device. This adapts them for general service, and specially for drilling holes at a steep upward angle, as required in overhand stoping and in making "raises." Usually the automatic feed consists of a light telescopic standard, screwed into the back of the cylinder. It is supplied with compressed air, which keeps the drill fed up to its work as the hole is deepened. Incidentally this device furnishes an air cushion, relieving the operator from much of the annoyance caused by vibration. These machines may be mounted on a light column, when desired, for breast-stoping, drifting, and similar work. Details of the air-feed are given hereafter.

Leyner Drill. Although this well-known machine must be classed with the "hammer" drills, its construction and operation are widely different from the numerous small, light machines working on the hammer principle. It occupies the same field as the standard drills of the reciprocating type; is designed for mounting on tripod or column and is provided with automatic rotation of the bit.

The most important drill of this make is commonly designated as the "Water" Leyner. Fig. 146 shows the longitudinal section. The cylinder, 17, is carried in guides in the shell, 1, and is fed by the feed screw and nut, 23 and 21, and feed crank, 26; 13 is the hammer, 7 the chuck, in which the drill shank is held by the chuck-key, 6; the chuck parts, including the buffer, 3, being retained in place by the cap, 2. The air valve, 15, is of the spool or piston type, controlling the main ports, and being itself actuated by a system of small auxiliary ports, opened and closed by the reciprocations of the hammer.

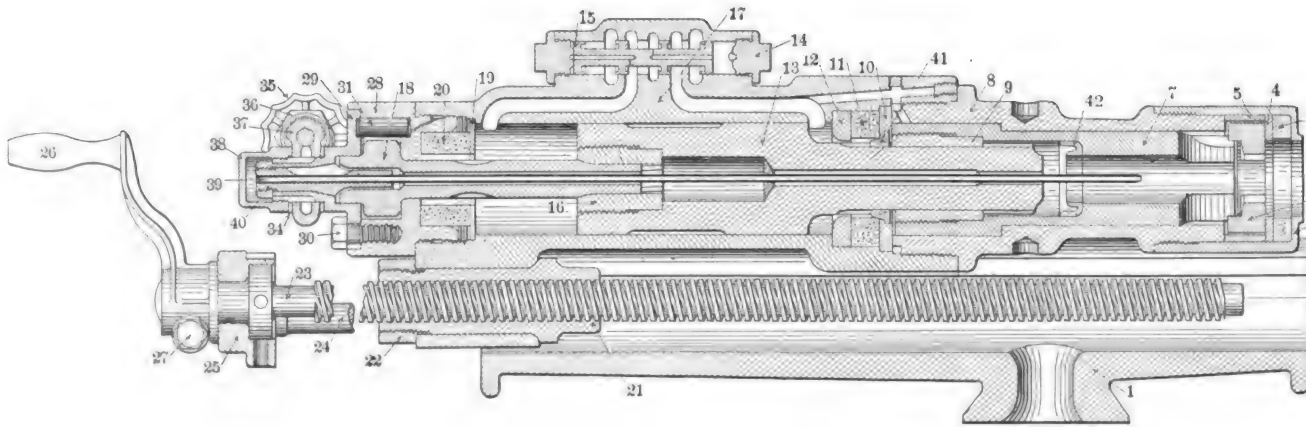


FIG. 146.—"Water" Leyner Drill.

Rotation of the bit is effected as follows: The rifle-bar, 18, which is controlled by the pawl, with spring and plunger, 31, 32, and 33, engages with the rifle-nut, 16, screwed into the hollow rear-end of the piston or hammer. This causes rotation of the hammer on each back stroke. The forward, smaller end of the hammer is fluted, and engages with an internally fluted bronze nut, 9, in the rear end of the chuck. Thus the chuck, holding the bit, is caused to rotate with the hammer. These parts are shown separately in Fig. 147. The drill bit, also shown in this

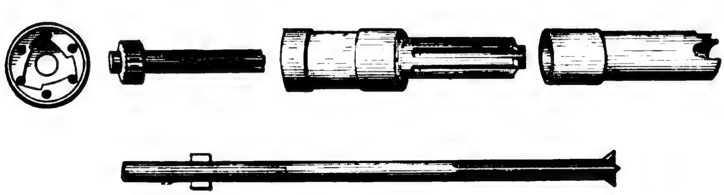


FIG. 147.—Rotation Device of "Water" Leyner Drill.

cut, is of special form, being not only hollow, for the passage of the water into the drill hole, but provided also with two lugs, by which it is locked in the chuck.

The water supply, already referred to, is furnished under pressure from an 18-gallon steel tank, accompanying the drill and connected to it, at 37, by a length of hose. Another hose conveys compressed air from the main to the tank. Water is thus forced from the tank through the water tube, 39, which passes through the rifle-bar and hammer, in the axis of the machine, and is delivered into the hollow bit, when the latter is locked in the chuck ready for work. A back buffer, for easing the shock, if the hammer should overrun its stroke, is shown at 20, with its plate, 19; 11 and 12 are the front cylinder buffer and buffer ring. At 41 and 42 are leather washers for making air-tight the joints between chuck and hammer.

The use of the water jet undoubtedly increases the rate of drilling; it keeps the hole clean, so that the bit is not cushioned —as it may be with other machines—by an accumulation of stiff sludge in the bottom of the hole. In solid rock of uniform texture

the Leyner does good work, but is not so successful in fissured, seamy formations, where a machine drill bit is likely to stick or "fitcher." Moreover, to obtain the best results from this machine, more careful and intelligent handling is required than for most of the reciprocating drills.

The "Water" Leyner drill is made in two sizes. No. 2 has a 2½-inch cylinder and 9-lb. hammer, the drill head, bare, weighing 130 lbs., or, with all fittings, boxed for shipment, 205 lbs. No. 3 drill, bare, having a 3-inch cylinder and 13½-lb. hammer, weighs 155 lbs.; including fittings, 230 lbs. Without the leg-weights, the tripod weighs about 200 lbs.; 6½-ft. column, 275 lbs., and water tank, 70 lbs.

Another Leyner drill is the "Rock Terrier," a small machine for stoping and similar work, handled by one man. It is made in two patterns: the "dry," of which the drill head, bare, weighs 52 lbs., and the "water" pattern, bare, 54 lbs. The hammer weighs 2¾ lbs.; 6-foot column, with clamps, 85 lbs., and tripod with shell, but without leg-weights, 46 lbs.

Still another pattern of the Leyner drill is the No. 5, a light-weight but strongly built machine, specially designed for stoping. It differs essentially from the larger machines of the same make: (1) in having no rifle-bar, with its accessories, for the automatic rotation of the bit; (2) in being provided with an automatic air-feed cylinder or standard, such as is found in a number of the machines described in the following pages. The valve-chest, containing a spool-valve somewhat similar to that of the "Water" Leyner, is attached to the side of the main cylinder.

Hardsocg Wonder Drill. Fig. 148 is a longitudinal section of one of the smaller sizes, provided with a D-handle; Fig. 149 shows a drill with the air-feed attachment.

The operation of the hand drill will be understood by referring to the first cut, in which the hammer is shown at the end of its forward stroke. Compressed air is admitted at the nipple, 2, to which is attached the air valve and hose connection to the main. From 2 the air enters the annular recess, 3, and acts constantly on the shoulder, 13, of the hammer. At the beginning of the forward

stroke, the 2-way ports, 5 and 6, passing through the head of the hammer, are opposite the recess, 3. Air is thus admitted into and behind the hollow hammer, and the area presented to the air pressure being much greater than the area of the shoulder, 13, the forward stroke is made. Just before the hammer strikes the bit, the ports 5 and 6 reach the annular space 8, in the front end of the cylinder. The air is exhausted into 8, whence part of it is discharged directly into the atmosphere by the exhaust port, 4, the remainder passing through the hollow bit into the bottom of the drill hole. (When drilling in soft rock, and especially when the presence of moisture produces a pasty sludge, all of the exhaust

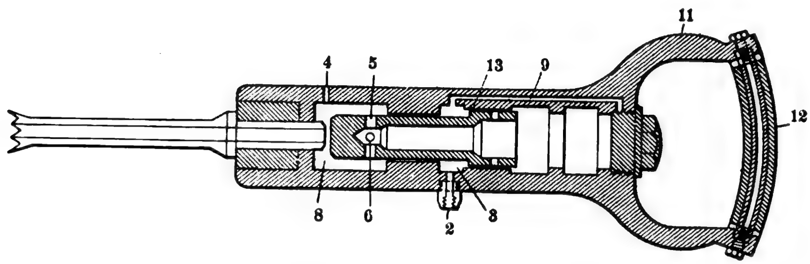


FIG. 148.—Hardsocg Wonder Drill, with D-Handle.

air may advantageously be discharged through the bit, by plugging the port, 4.) The exhaust having taken place, the back stroke is made by the constant pressure of the inlet air on the annular shoulder, 13, of the hammer. This drives back the hammer until the ports 5 and 6 are again brought opposite the recess 3, thereby admitting air behind the hammer.

The drill is fed forward by hand, and is rotated by the handle, 12, the bit shank being octagonal. To take up vibration, the handle is made of rubber tubing, held in place by studs, as shown. An air-feed cylinder (see second cut), may be screwed into the rear end of the drill cylinder, by removing the large plug and the rubber handle. The rotation is then effected by manipulating the handles, 11. When using the air-feed, the admission port, 2, is plugged, the air entering from the feed cylinder through the

longitudinal port, 9. These drills strike from 1,800 to 2,000 blows per minute, at 90 to 100 lbs. air pressure.

Fig. 149 is a section of one of the larger machines, provided with the air-feed and designed for drilling up-holes. The feed cylinder, which is from 3 to 4 ft. long, is "broken" in the cut, to reduce the length. When at work, the drill rests on the pointed end of the feed cylinder, as nearly as possible under the place where the hole is to be drilled, and the operator steadies and holds the machine in the exact position required. Holes at a considerable angle to the vertical may thus be readily drilled.

Air is admitted at 15. A small portion of it passes through the port 29 to the rear of the piston, 17, by which the machine is fed forward automatically. Air for the drill cylinder passes from

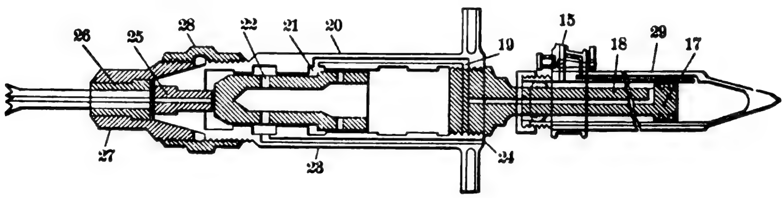


FIG. 149.—Hardsocg Air-Feed Stoping Drill.

15, round the feed piston rod, to the central port, 18; and thence, through ports 19 and 20, to the annular recess, 21, surrounding the hammer. When the transverse port, 22, comes opposite to the recess, 21, air is admitted to the hollow hammer, causing the forward stroke. At the end of the stroke the exhaust takes place by the ports 22 and 23, opening to the atmosphere at 24. As in the hand drill, the back stroke is produced by the constant pressure of live air on the annular shoulder of the hammer.

The impact of the hammer is received by the anvil or tappet block, 25. The front head, 27, in which is set the chuck, 26, is attached to the cylinder by the sleeve, 28. This sleeve has differential threads, for locking it firmly in position.

The hand drills, like that in Fig. 148, are made in four sizes, weighing from about 19 to 45 lbs. Air-feed machines (Fig. 149),

in a number of sizes and patterns, weigh unmounted from 35 to 65 lbs.; or, including their light columns and arms, from 95 to 160 lbs. In quarry work, for making straight rows of holes, as in getting out dimension stones, air hammer drills may be mounted on a horizontal bar, supported at each end by a pair of adjustable pointed legs. This mounting is similar to, though much lighter

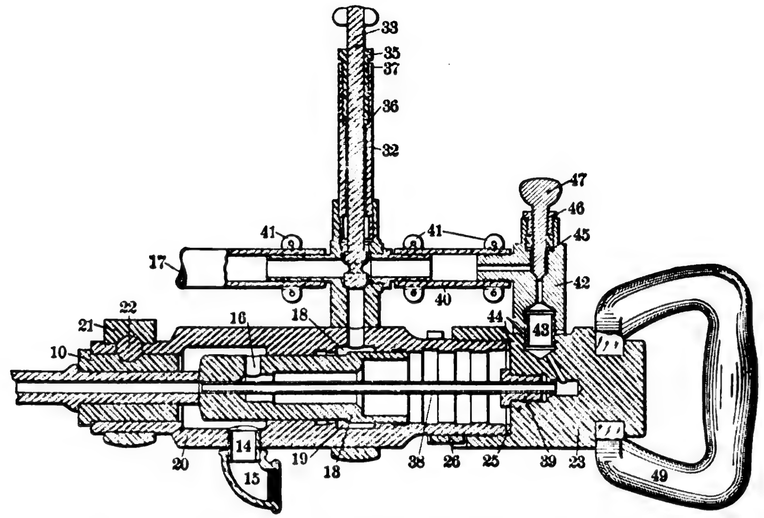


FIG. 150.—Murphy Hammer Drill, with D-Handle, for Sinking.

than, that used in quarry service for ordinary reciprocating rock drills.

Murphy Drill. This resembles the Hardsocg in general construction, though with differences in details.

Fig. 150 illustrates the design of the cylinder and ports of the latest model of the "sinker" drill, provided with D-handle. The hammer, 13, is bored out from the rear end, and has a transverse port, 16, near its forward end. Compressed air enters at the hose connection 17 and, on raising the throttle valve, 33, passes to the annular recess, 18, in the cylinder 20. The throttle is set in a branch pipe, 32, screwed into the cylinder, and has a gland, 35, lock nut, 37, packing and washer, 36. (Incidentally this valve

casing serves as a second handle, assisting the operator in supporting and rotating the machine.) On entering the recess, 18, the air acts on the shoulder, 19, thus driving back the hammer. At the end of the back stroke, the port, 16, is in connection with the recess, 18, and compressed air is admitted to the hollow hammer. The forward stroke is thus made, as in the Hardsocg drill. When the port, 16, reaches the larger diameter of the cylinder, at the end of the forward stroke, the air at the rear of the hammer is exhausted through the port, 14. The stroke is then reversed by the constant pressure on the shoulder, 19. The back-head, 23, of the cylinder is screwed on like a cap. It has a lock-key 26, and a washer, 25, which serves also as a buffer. A simple bushing, 10, holds the bit, which has a collar as shown, but neither chuck-clamp nor bolts are used. This bushing is keyed in place by the bolt, 22, in the ring, 21, encircling the forward end of the cylinder.

Live air is admitted to the hollow drill steel, through the hose connection, 40, from the throttle valve to the auxiliary valve, 47. This is set, with gland and washer, in the valve casing, 42, attached to the back cylinder head by the nipple, 43. From the auxiliary valve the air passes through a small diagonal port to the rear end of the tube, 38, which is held in the back head by the gland, 39, and communicates with the hollow steel through the forward end of the hammer. By manipulating the auxiliary valve, the operator regulates as required the quantity of air passing to the drill hole.

The Murphy sinker drill is made in two sizes, Nos. 3 and 5, weighing 55 and 65 lbs., and using respectively about 50 and 60 cu. ft. of free air per minute.

The air-feed stoping drill, for overhead work, is built in two models, two sizes of each; weighing, without column and arm, from 85 to 95 lbs. Model L is shown in Fig. 151. Air enters by the hose connection, 17; passing thence through the passages, 18 and 19, to the drill cylinder. Admission is controlled by the hand valves, 7, 7. From 18 a small quantity of air is admitted through port, 16, to the upper end of the feed cylinder, causing the inner tube, 50, carrying the pointed end, 61, to slide outward. Part of the feed air passes through the central port, 35, to the

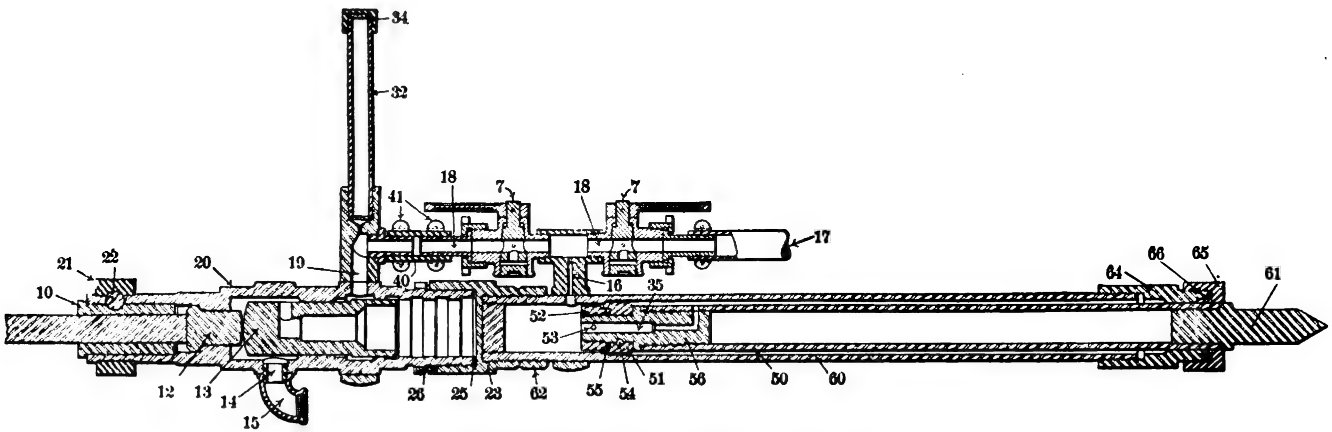


FIG. 151.—Murphy Air-Feed Hammer Drill.

annular space between the inner and outer tubes, and acts on the shoulders shown at each end. The standard is thus lengthened, feeding the drill forward automatically as the hole is deepened, and keeping the bit firmly pressed against the bottom of the hole. The stuffing box for the telescoping tubes is shown at 64, 65 and 66.

In this machine the branch pipe, 32, instead of containing the throttle, is used merely as a handle for rotating the drill. The hammer does not strike the bit directly, but delivers its blows on the tappet block, 12. In other respects the cylinder and its appurtenances are substantially the same as those of the Murphy sinker drill, described above. Owing to the mode of construction of this model, the air hose connection turns with the machine, so that in rotating it the operator cannot make complete revolutions, but must turn it back and forth. Care must be taken to set up the drill so that no dirt will enter the lower joint of the stuffing-box. This joint, however, is protected by its position from the cuttings falling from the mouth of the drill hole.

The other model, H, of the stoping drill has the hose connection attached to the outer tube of the feed cylinder, and the stuffing-box is at the end nearest the drill cylinder. In this position the stuffing-box is more exposed to wear from the drill cuttings, but as it is not near the floor the machine can be set up anywhere, without a supporting block or plank. Moreover, in rotating the drill the operator can make complete revolutions, if desired.

Sullivan Hammer Drills. Unlike the machines already described these have a valve for distributing the air. Figs. 152 and 153 show longitudinal sections of the latest pattern of hand drill, with hollow bit, and illustrate the valve motion and arrangement of ports, which are essentially the same for all the models. In the earlier forms a solid spool-valve was used; in the present design the valve is cup-shaped and works on the spindle, Z, which is a part of the valve-box cap, as shown by the cuts. The section in Fig. 153 is taken at about 45° from that in Fig. 152, for the purpose of exhibiting all the longitudinal ports.

Air is admitted, through the hose connection to the port Y

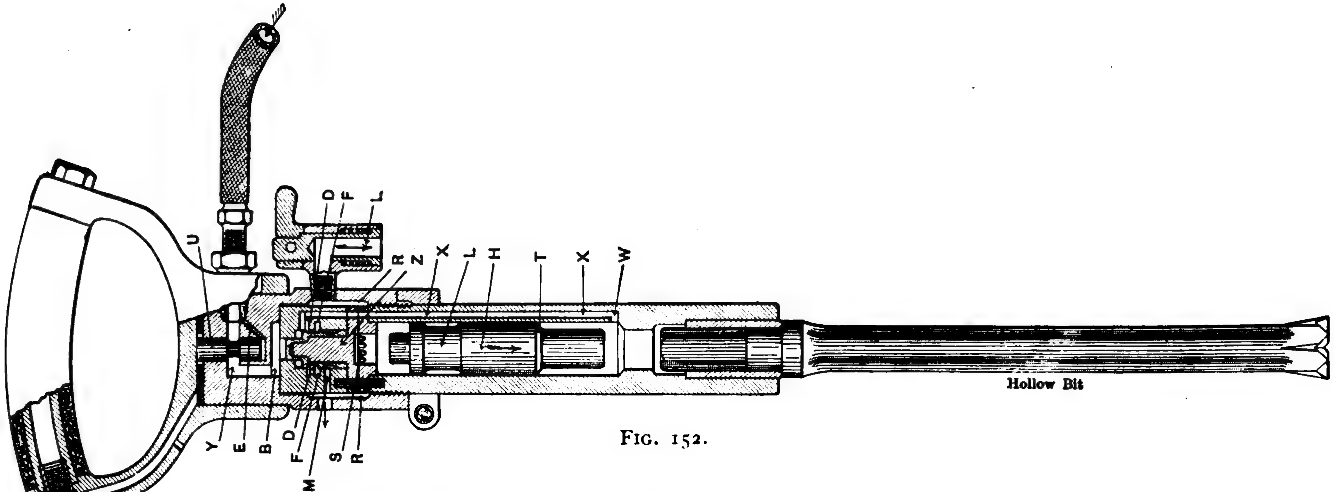


FIG. 152.

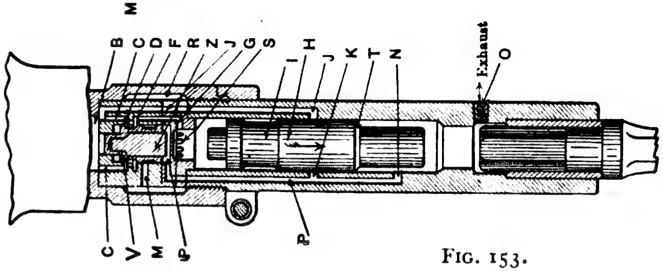


FIG. 153.

Sullivan Hammer Drill, for Sinking and "Plug" Holes.

and recess, B, simply by pressing the handle of the drill. This forces down the throttle plug, U, to the position shown. When the handle is released, the plug rises automatically, under constant pressure from the small live-air lead, E, and closes the port Y. The hammer, H, is shown on its forward stroke. Air enters from B, through several longitudinal passages, C, to the annular port, D; thence, by the rear annular valve groove, to port F, and finally, by a series of other longitudinal passages, G, through the ports, S, to the cylinder.

While the drill is in operation, the port, J, communicating with B, is occupied by air at constant pressure. Previous to the delivery of the blow of the hammer, its groove, I, comes opposite to the port, J, thereby putting the port, K, in communication with J, and admitting live air through P to the forward or annular end of the valve, V. As the area of this end exceeds that of the other (which is subject to constant air pressure from the inlet B), the valve is reversed. This puts the annular valve port or groove, D, in communication with the long reverse port, X W. Meanwhile, the hammer completes its stroke on momentum, while air is being admitted at W to the forward end of the cylinder, preparatory to reversing. The air in the rear end of the cylinder, which caused the forward stroke, exhausts through the series of ports, S, leading to the corresponding longitudinal passages, G, thence to the annular valve port, F, and finally reaches the atmosphere through M.

The back stroke is caused by the pressure of the air on the annular shoulder, T, of the hammer. When the small, forward end of the hammer passes out of the contracted portion of the cylinder, the air which was used to hold the valve upward passes through port N to the exhaust opening, O. Simultaneously, the air which caused the back stroke also exhausts through O. The live air in B now acts on the upper or smaller end of the valve, driving it forward and reversing the hammer.

This type of Sullivan drill is made in two sizes. The "plug" drill, for light work, has a 1-5/16-inch cylinder and weighs 18 lbs. Either hollow or solid steel may be used; if solid, part of the air

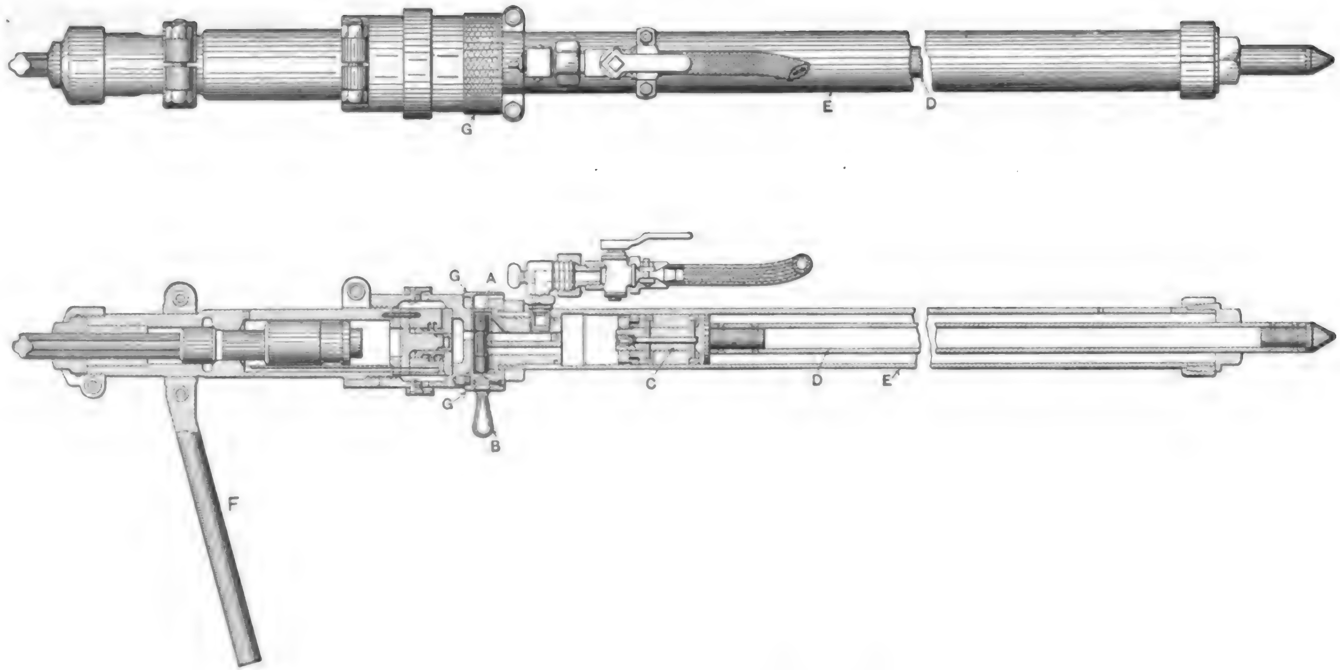


FIG. 154.—Sullivan Air-Feed Hammer Drill.

is exhausted through a hose connection to an annular rider on the bit shank, by which jets of air are directed into the hole. Another size, with 1¾-inch cylinder, weighs 30 lbs., uses hollow steel, and bores holes up to 1⅝-inch diameter, with an air consumption (at 100 lbs. pressure), of about 25 cu. ft. free air per minute.

The air-feed drill is shown in Fig. 154. The valve motion is the same as described above, except in the arrangement of the air admission passages and the position and manipulation of the throttle, A. The latter is set transversely in an eccentric sleeve, G. As the sleeve is turned by the handle, around the drill, it carries with it the valve. The eccentric inner surface of G causes the valve to take the different positions necessary for controlling the admission ports. First, it opens the port to the feed cylinder, E, and then, by turning the handle farther, opens the port leading to the constant pressure chamber back of the main valve, as in the hand drill. The throttle plug is held against the eccentric surface of the sleeve by the small lead of live air, as shown.

The feed cylinder, E, is similar to those of the other makes of air-feed drills, already described. Live air from the admission port acts on the piston, C, which is attached to the inner telescoping tube, D. When in operation, the whole drill is rotated back and forth by the handle, F. The diameter of cylinder is 2 inches; weight of machine, 70 lbs.; maximum diameter of hole drilled, 2 inches; free air consumption (at 100 lbs. pressure), 35 to 40 cu. ft. per minute.

Ingersoll-Rand "Imperial" Hammer Drills. This company makes hammer drills of two distinct classes, namely: the "Imperial," which is valveless, and the "Crown" drill, provided with a spool-valve. Each of these classes comprises machines of several different sizes and weights, both of the D-handle and telescope-feed types.

The "Imperial" hammer drill, with D-handle, types M V-1 and M V-2, is shown in longitudinal section by Fig. 155. Its general construction is similar to that of the valveless machines already described. Air is admitted at the connection, 13, into which is screwed the nipple, 46, shown in the figure in the lower

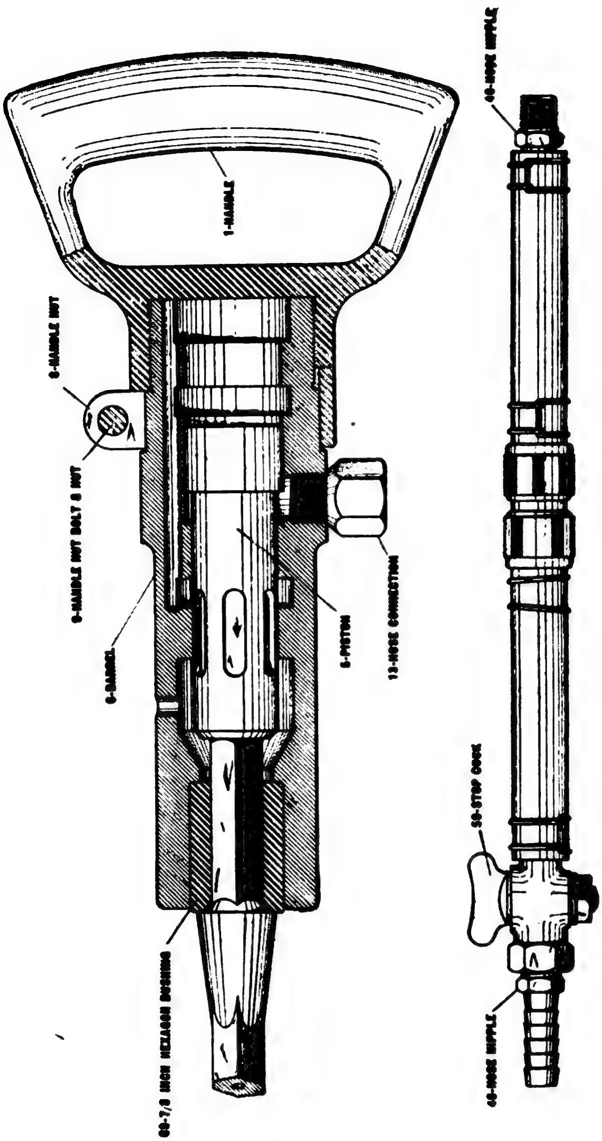


FIG. 155.—Ingersoll-Rand "Imperial" Hammer Drill. Types M V-1 and M V-2.

part of the cut. To 13 is attached a short hose connection, with stop-cock, 59, and nipple, 48, for joining to the hose leading from the air main. The hollow piston 5 has a shoulder near the rear end, against which the air pressure is constantly acting, and a series of 6 slot-shaped ports near the forward end. In the cut the piston is shown as having completed its stroke, in which position the air is being exhausted through the piston ports and thence to the atmosphere. The return stroke is caused by the constant pressure on the shoulder of the piston.

Rotation of the drill is provided for by a straight handle, screwed or clamped by a gland to the cylinder. In this design the clamp bolt and nut are shown at 8 and 9. The shank of the bit is held in a bushing, 69, which is made in two forms, to receive either hexagonal or cruciform steel. Solid steel is used for these machines, which are designed specially for shallow "plug holes," both for mining and quarry service. They may be employed also for miscellaneous drilling, in connection with many engineering and contracting operations.

Fig. 156 illustrates the "Imperial" air- or telescope-feed hammer drill, type M C-12. The hammer, 5, is shaped differently from that of the M V style, described above, but its action is similar. An anvil-block, 22, transmits the blow of the hammer to the bit shank. The bit bushing, 67, is held in the cylinder head by the cottar-bolt, 86. The construction of the telescope-feed is clearly indicated in the cut and will be understood by reference to the descriptions already given of air-feed hammer drills. Compressed air is admitted to the machine at the connection, 36, controlled by an air cock. From the upper end of the telescope feed tube the air passes through several radial ports in the rear cylinder head, and thence into the cylinder through longitudinal ports, as shown. For rotating the machine a straight handle, 48, is screwed into the cylinder casting.

The "Imperial" drills weigh from 42 to 65 lbs., unmounted; approximate air consumption, at pressures from 60 to 100 lbs., 13 to 57 cu. ft. per minute. Travel of telescope-feed, 20 inches; total length of drill, with feed run in, 48½ and 50 inches. Number

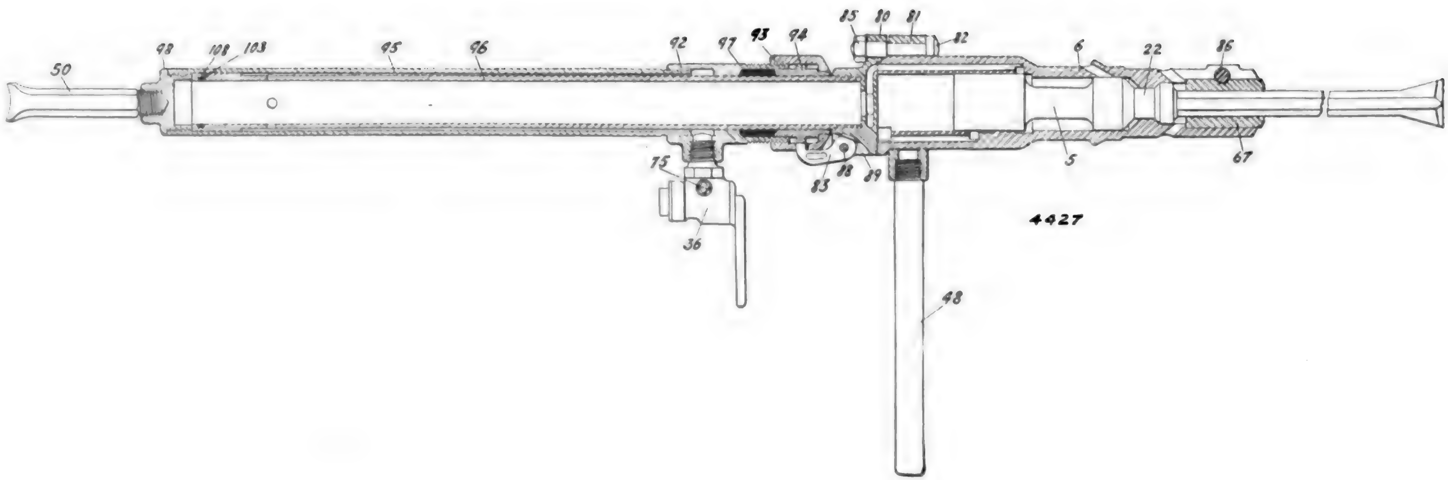


Fig. 156.—Ingersoll-Rand "Imperial" Hammer Drill, Type M C-12.

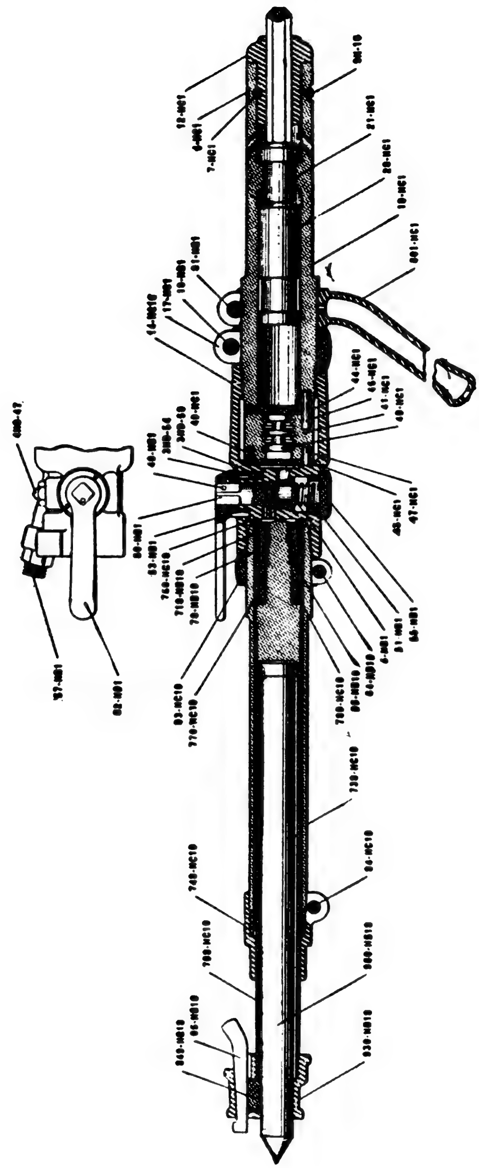


FIG. 157.—Ingersoll-Rand "Crown" Hammer Drill, with Air-Feed, Type H C.

of blows per minute, at 60 lbs. pressure, from 990 to 1,320; at 100 lbs., from 1,120 to 1,560.

Ingersoll-Rand "Crown" Hammer Drills. These machines are provided with spool-valves and are known as types H A, H B, and H C, according to size and weight. General longitudinal sections of the telescope air-feed and D-handle styles are shown in Figs. 157 and 158.

The valve motion is illustrated by the diagrams, Figs. 159 and 160. At the beginning of the stroke the valve, F, and hammer, H, are in the positions indicated in Fig. 160. Air enters port A from the hose connection. The inner end of A being now closed by the hammer, the air passes through ports B and C to the valve chest. Part of the air entering through C escapes by the small port, D, thus reducing the pressure in the forward end of the valve chest, E, below the working pressure. The pressure in E holds the valve in the position shown, so that the air entering at B passes around the valve into the port G, and thence into the rear end of the cylinder. This drives the hammer forward. While the hammer is making its stroke, part of the air in the front end of the cylinder exhausts through port I, and part by the longitudinal port, J, around the valve, F, and finally through port K to the atmosphere.

Referring to Fig. 159: as the hammer approaches the end of its forward stroke, the annular groove, L, is opposite port M, thus allowing live air from port A to pass into the longitudinal port, O, and thence to the rear end, P, of the valve chest. The pressure of this live air being greater than the reduced pressure in chamber E (Fig. 160), already referred to, the valve is thrown forward to the position shown in Fig. 159. Then air from port B passes around the valve and through port J, to the front end of the cylinder, thus causing the back stroke of the hammer. During this stroke the air in the rear end of the cylinder exhausts through port G, and thence around the valve to the exhaust port Q. While on its back stroke the hammer opens port I, allowing all compressed air in port O, and in the front end of the cylinder, to escape. This exhausts the air from the chamber, P, of the valve chest, and the

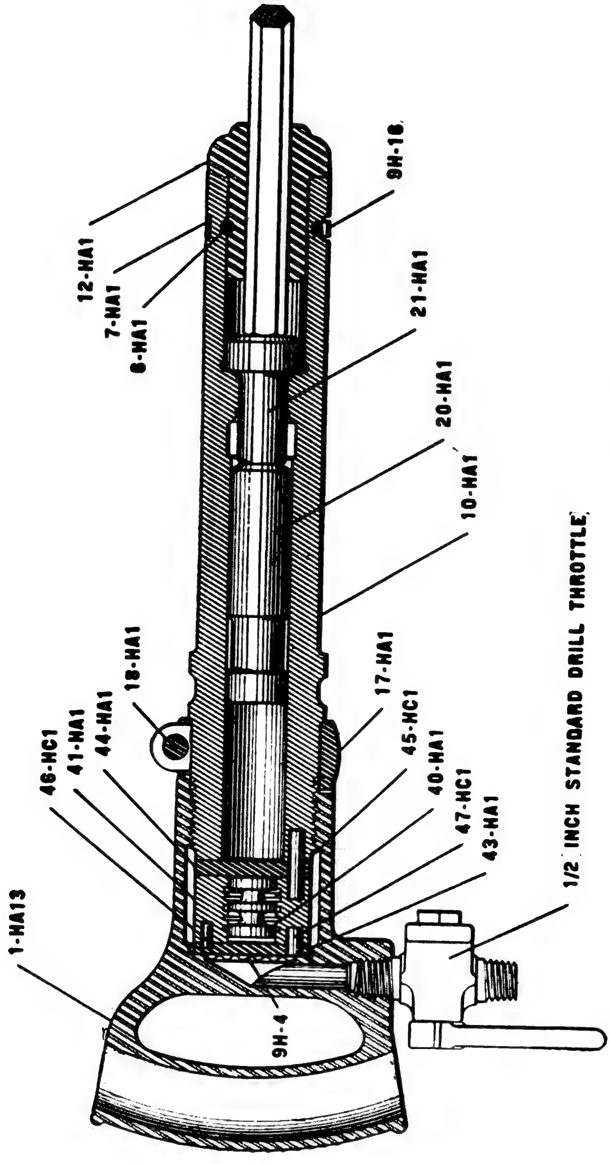
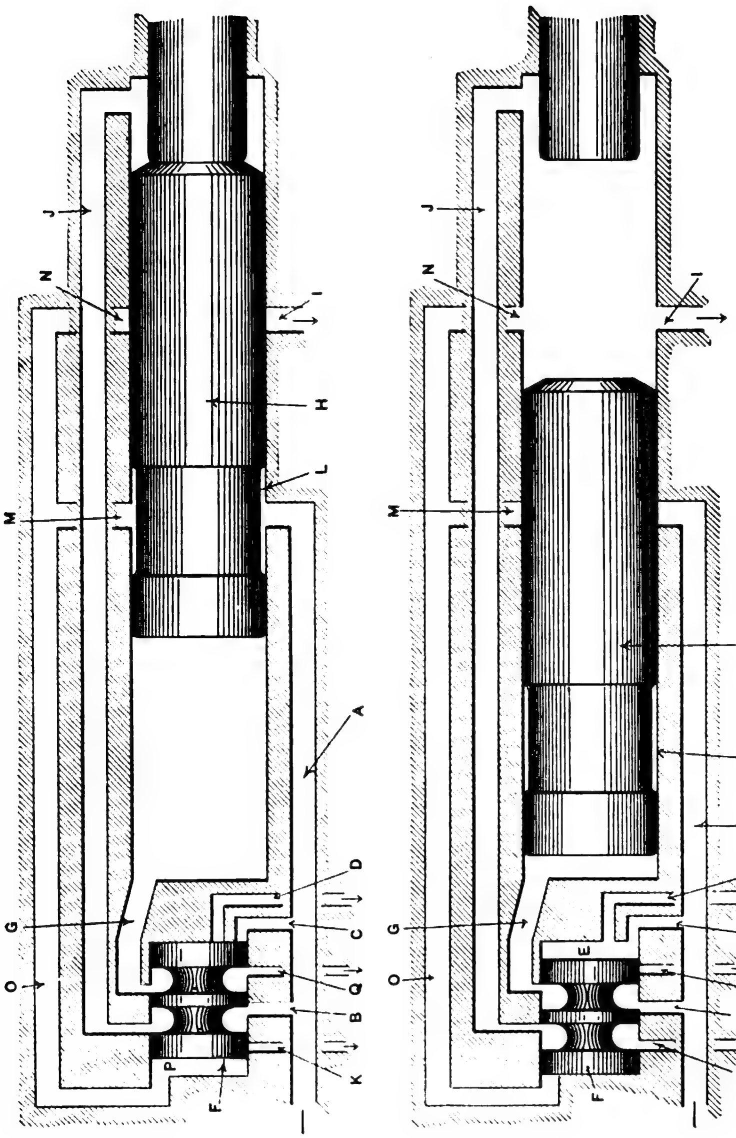


FIG. 158.—Ingersoll-Rand "Crown" Hammer Drill, with D-Handle, Type H A.



FIGS. 159 and 160.—Ingersoll "Crown" Hammer Drill; Types H B and H C. Diagrams Showing Valve-Motion and Relations of Ports.

constant reduced pressure at the other end of the chest (E, Fig. 160) causes the valve to reverse and assume its original position, thus completing the cycle of operation. The H A-14 type of this drill is provided with a short hose, from the valve chest to a "dust-hose nozzle" encircling the bit at the mouth of the drill hole.

The telescope-feed machines weigh from 40 to 80 lbs., unmounted, and consume, according to the makers, from 18 to 60 cu. ft. of free air per minute, at working pressure from 50 to 100 lbs. The travel of the telescope-feed is from 18 to 24 inches; total length of drill, with feed run in, from 50 to 59 inches. Number of blows per minute, at 60 lbs. pressure, from 1,050 to 1,200; at 100 lbs., from 1,300 to 1,400.

Waugh Hammer Drill. Fig. 161 shows in plan and longitudinal section type 8-C, of this drill, provided with air-feed standard for use in stoping.

Admission of air is controlled by the taper throttle valve and handle 12, the valve being in its open position. From the hose connection 13, the air follows the path indicated by the arrows drawn in full lines, through the hollow throttle to the head of the feed cylinder; thence back through a transverse port in the throttle, and through longitudinal passages in the valve chest 15, to the annular groove A in the chest. The valve 16 has a differential shoulder, against which the live air in groove A acts, thus throwing the valve back into the position shown in the cut. In this position the air passes the forward edge of the valve into the main cylinder, and drives the piston or hammer forward against the tappet block, 8. During the stroke, after the hammer passes port G, the air in front of the hammer is exhausted through port F and passages E and D to the annular groove C in the valve chest; and thence past the valve through ports H H and passage K to the atmosphere, as shown by the broken line arrows. At the same time, as soon as the hammer uncovers the port P, live air flows from the cylinder through P (and the small dotted port forming its continuation), to the back or larger end of the valve, 16, thus throwing the valve forward. This shuts off admission of air and opens the cylinder space behind the hammer to the

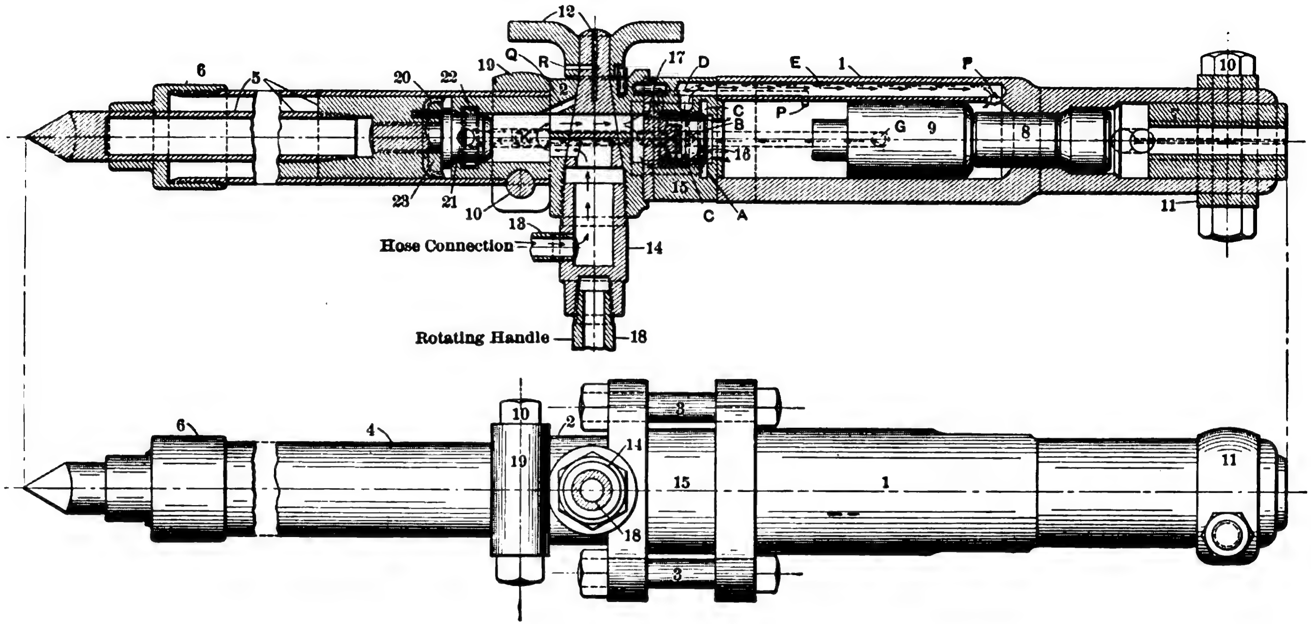


FIG. 161—Waugh Air-Feed Hammer Drill.

exhaust, preparatory for the return stroke. The exhaust air passes through two round dotted ports, H H, nearly opposite the rear end of the valve, to the longitudinal passage K and thence to the atmosphere. The exhaust is sufficiently throttled down in passing through H H, to keep the valve 16 in its forward position by the pressure exerted on its rear or large end.

In this position of the valve, live air passes as indicated by the full line arrows, from the annular groove A, through the ports B in the valve seat, to the groove C in the valve chest, and thence by passages D and E and the port F, to the forward end of the hammer, thus causing the return stroke. When, on the back stroke, the hammer uncovers the port G, the air exhausts from the front end of the cylinder through G and thence to the atmosphere by the passage K, as indicated by the broken line arrows. On completion of the back stroke, the exhaust pressure acting on the larger end of the valve 16 has fallen; so that the live-air pressure in the groove A, acting on the small end, throws the valve back to the original position (as in the cut). Towards the end of the stroke the small end of the hammer enters the hollow valve, and there confines and compresses a small quantity of air, thus accelerating the back throw of the valve.

It may be added that the exhaust air goes through the longitudinal passage K, and so does not disturb the dust falling from the drill hole.

When the throttle is closed, the inside of the feed cylinder 4 is open to the atmosphere through passages Q and R. When the throttle is open, the feed cylinder is closed to the atmosphere, and at the same time live air is admitted to the feed cylinder through a small port in the throttle (not shown), and the drill is carried forward to its work. At the same time air enters from the hose connection through the hollow throttle and acts on the piston of the feed cylinder.

The stoping drill is made in two sizes, weighing respectively 50 and 60 lbs., and using solid bits. Drifting drills (8-D and 3-D), weighing 75 and 60 lbs., are designed for a column mounting and may be used also for sinking. For these two machines, hexagon,

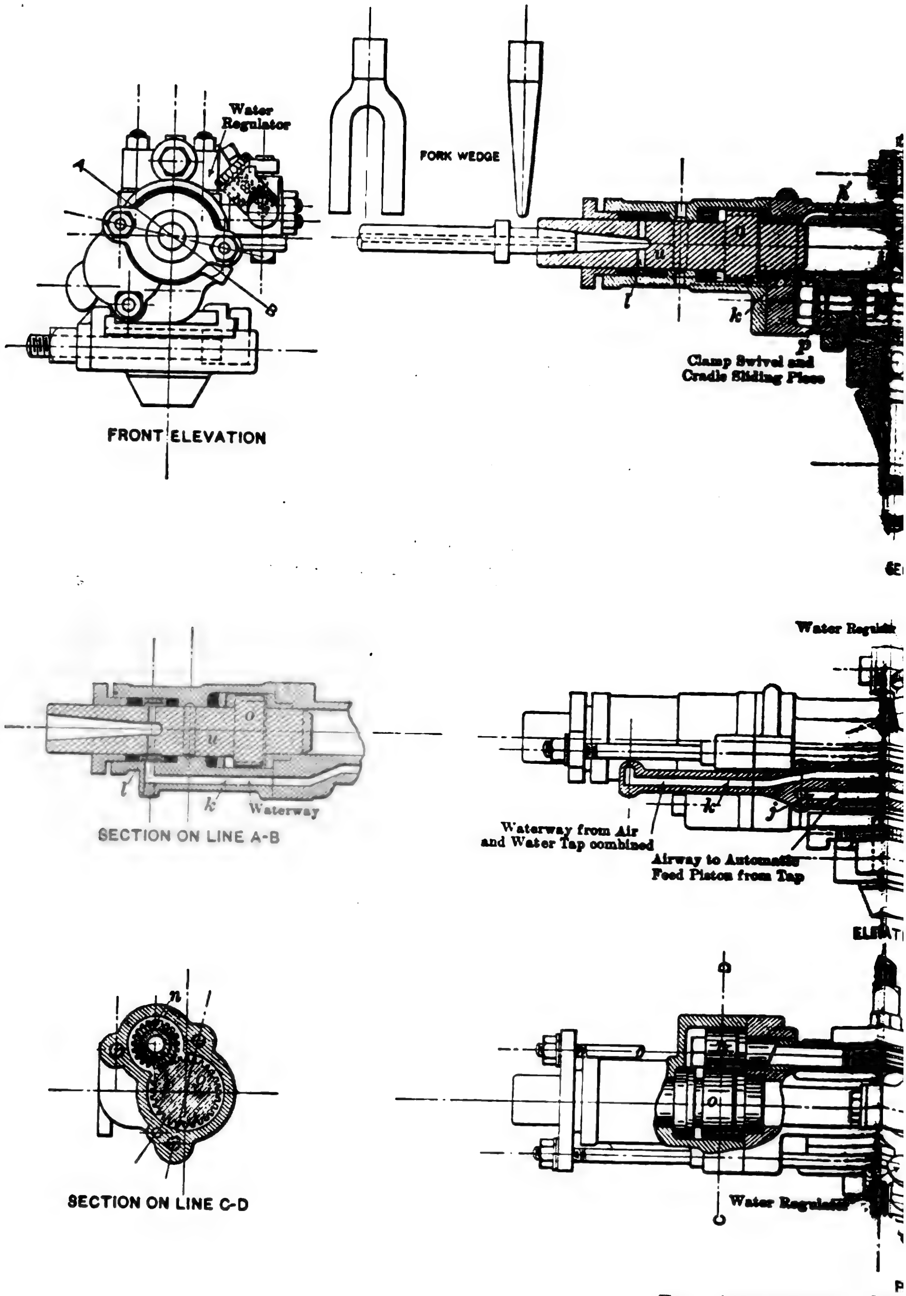
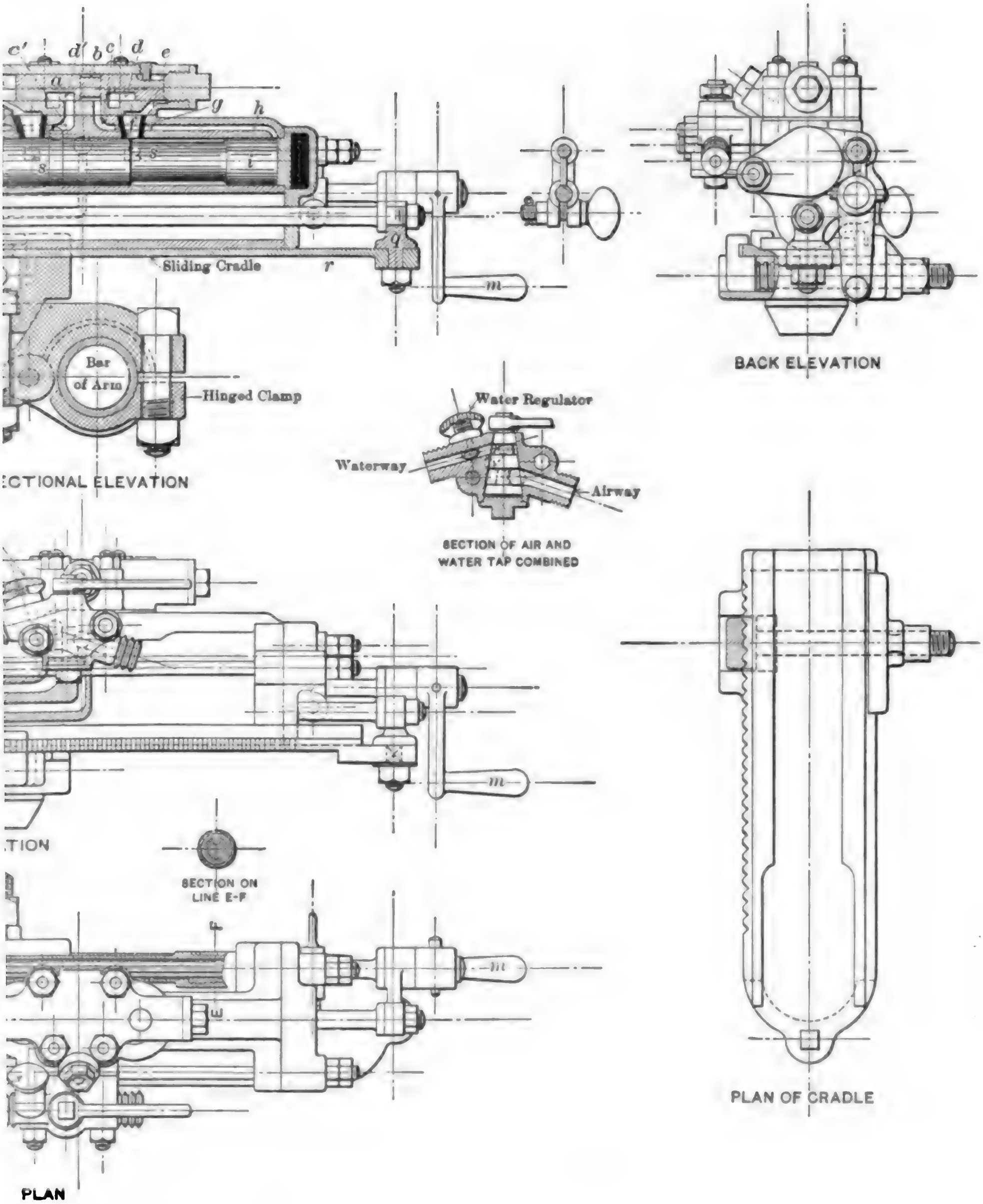


FIG. 162.—Stephens' "Clim



imax Imperial" Hammer Drill.

hollow steel is used, through which live air or water is delivered by means of a 3-way valve. This valve is screwed into the side of the drill head, and is opened or closed by a sleeve fitting over the rotating handle, 18, so as to be readily manipulated by the operator. Water is supplied from a small tank, under pressure of compressed air, similar to that described under the Leyner drill.

Small hand machines, with D-handle, for sinking and for drilling "plug" holes, are also furnished by the same makers. They weigh respectively 48 and 27 lbs.

Stephens' "Climax Imperial" Hammer Drill; an English stoping drill, made at Carn Brea, Cornwall. It has a $1\frac{3}{4}$-inch cylinder, weighs 75 lbs., and is intended to be mounted on a light column or bar. In several features of its design the Climax drill is in marked contrast to the American hammer drills.

Referring to Fig. 162, it will be seen that the valve motion resembles that of the Climax reciprocating drill, described in Chap. XX. Air is admitted by the combined air and water tap (detailed section), attached to the side of the valve chest; thence passing by the annular recess *b*, in the piston valve *a*, through *c* and *c'*, to the main cylinder ports *h*, *h'*. As shown by the cut, the valve and hammer are in position to begin the forward stroke. The recesses *d*, *d'*, in the valve, communicate with the main exhaust (not shown). Air is constantly admitted to both ends of the chest by a very small groove *t*, the valve being thrown by exhausting through the much larger auxiliary ports *e* and *f*. When the drill is in operation, the ports *f* are alternately brought into communication with the annular recess in the hammer, thus releasing the air, by way of the square ports *s* and *s'*, to the main exhaust. The ports *f* are lined with hollow, conical plugs, *g*, of composition metal, which are shaped below to the curve of the hammer. They are kept in close contact with the hammer, for preventing leakage of air, by the pressure of the valve-chest, when bolted in place. When the plugs wear too loose, a thin washer is inserted above them.

The water for the drill hole is best supplied by gravity, under a pressure of say 15 lbs. It enters the combined air and water tap, or throttle, already noted, through the passage *k*, to the transverse

port *l*, in the cylindrical anvil block *u*, which serves as the drill holder, or chuck. Thence the water passes to the hollow bit (see the elevation and the "section on line A B"). The drill may also be used, under proper conditions, for "dry" holes, the dust being allayed by an external spray from the throttle.*

The machine has an automatic air-feed. A small piston, *p*, with its rod, is rigidly bolted to the lug *q*, on the cradle or shell *r*. Air from the throttle passes through the passage *j* to the feed cylinder, thus forcing the entire drill head forward on the shell and keeping the bit pressed against the bottom of the hole. After the machine has been fed forward 14 inches (the working length of feed), the air is shut off and the transverse bolt, shown in the plan of the cradle, is slacked. The operator then slides the cradle forward on its support under the machine, tightens up the bolt on the serrated edge of the cradle, and proceeds with the drilling. Thus, a total feed of twice the length of the cradle—or about 28 inches—is obtained without putting in a longer bit.

Rotation of the bit is effected by hand. The bit is held merely by friction in the conical socket of the chuck. Gear teeth are cut on the periphery of an enlarged part, *o*, of the chuck, engaging with which is a smaller gear *n* (see general plan and the "section on line C D"), which is keyed on a spindle passing to the rear of the machine and rotated by the handle *m*.

Excellent drilling records have been made by this machine, both in England and South Africa.

Makers of Hammer Drills. The hammer drills chosen for illustrating the construction and operation of this type of machine are believed by the author to be among the best of their respective classes. Details of others, equally good, might be given if space permitted. A number of these machines are on the market; in fact, within the past few years, most of the manufacturers of reciprocating rock drills have added to their lists one or more styles and sizes of the hammer drill. Arranged alphabetically below are the names of most of the drills:

* This "dust allayer" is described in Chapter XX, under the Climax reciprocating drill.

Boyer ....(Chicago Pneumatic Tool Co., Chicago, Ill.).
Cleveland ....(Cleveland Pneumatic Tool Co., Cleveland, Ohio).
Climax ....(R. Stephens & Son, Carn Brea, Cornwall, England).
Flottmann ....(H. Flottmann & Co., Cardiff, Wales).
Franke ....(Made in Germany).
Iler ....(Iler Rock Drill Man'f'g Co., Denver, Colo.).
Ingersoll "Crown" (Ingersoll-Rand Co., New York).
Kimber ....(Formerly made in Johannesburg, S. Africa, now built by the Ingersoll-Rand Co., New York).
Leyner ....(J. Geo. Leyner Engineering Works Co., Denver, Colo.).
"Little Jap"........(Ingersoll-Rand Co., New York).
Murphy ....(C. T. Carnahan Man'f'g Co., Denver, Colo.).
Rand "Imperial" ..(Ingersoll-Rand Co., New York).
Schmucker ....(Great Western Pneumatic Tool Co., Denver, Colo.).
Shaw Eclipse ......(Shaw Pneumatic Tool Co., Denver, Colo.).
Sullivan............(Sullivan Machinery Co., Chicago, Ill.).
Waugh ....(Denver Rock Drill and Machinery Co., Denver, Colo.).
Whitcomb..........(Whitcomb Hammer Drill Co., Rochelle, Ill.).
Wonder.............(Hardsocg Wonder Drill Co., Ottumwa, Iowa).

Depth of Hole and Speed of Drilling. In connection with the work of hammer drills, the important questions of speed of drilling and the practical limit of depth of hole are closely related. Records of rapid work are often published and it has been stated, in general, that holes can be drilled at the rate of 3 to 4 inches per minute in rocks of the granite type, or from 7 to 9 inches per minute in limestone or ordinary sandstone. Such statements, though well substantiated, cannot be taken as general averages. They apply obviously to work done under favorable conditions. It is certain, also, that, with the air jet and to some extent also with the water jet, when the holes are "wet," *i.e.*, directed downward, they do not clean so well as depth increases. This begins to be noticeable at depths of even 2 or 3 feet, and the speed of drilling materially diminishes. Furthermore, as the length and consequent weight of the bit increases with depth of hole, the inertia also increases and the blows of the light hammer used in most of these machines become less and less effective. The deepest holes are made and the fastest work done when drilling "uppers" (holes directed at a steep upward angle), in dry rock or ore. The dust and cuttings will then run out freely by gravity; the hole is in effect self-cleaning and the blows of the hammer are more effective than when the

bit is clogged with cuttings, or, in wet rock, with a pasty sludge. In drilling dry "uppers," a depth of 5½ to 6 feet is probably near the economic limit for all except the largest sizes of the hammer drills.

Records of Work. The prefatory remarks, made under this heading in Chap. XX. respecting the work of reciprocating drills, apply also here. But, since hammer drills are almost invariably operated without mounting, no allowance of time for setting up the machines is necessary; and as there are no chuck-bolts to manipulate, changing bits usually takes only 1½ to 2 minutes for each bit.

Portland Gold Mine, Cripple Creek, Colo. Stoping in hard, phonolite breccia, with Leyner air-feed drill:

TABLE XLIV.

Hole.	Depth, Inches.	Total Time, Minutes.	Net Drilling Time, Minutes.	Inches Drilled per Minute.	
				Total Time.	Net Time.
1	24	25	16.5	0.96	1.45
2	37	13.5	11	2.74	3.36
3	17	7	6	2.43	2.82
Averages	26	15.2	11.2	2.04	2.54

In the same stope, working under the disadvantage of inconvenient, cramped positions, 8 holes, aggregating 21 ft. 7 in. deep, were drilled in 4 hours, including all delays, or at the average rate of 1.08 inches per minute.

At one of the *Michigan Copper Mines* the No. 5 Leyner, on a test run, drilled 159 holes, aggregating 1,141 feet in 39 shifts: or at an average rate of 29.3 feet per shift.*

Central Tunnel, Idaho Springs, Colo. An average of a number

* Extract from a letter from the J. Geo. Leyner Engineering Works Co., Dec. 15, 1909.

of holes in tough, hornblende schist, drilled by a "Water" Leyner machine, showed 2.64 inches per minute, including all delays.

Cresson Gold Mine, Cripple Creek, Colo. Stoping in hard, dense trachyte, containing numerous small quartz stringers; Waugh air-feed drill:

TABLE XLV.

Hole.	Depth, Inches.	Total Time, Minutes.	Net Drilling Time, Minutes.	Inches Drilled per Minute.	
				Total Time.	Net Time.
1	36	16	14	2.25	2.57
2	50	18	14	2.77	3.57
3	51	22	17	2.32	3.00
4	54	26	22	2.08	2.45
Averages	48	20.5	16.8	2.35	2.90

In this mine the average footage of hole per 8-hour shift is about 48 feet, or 1.2 inches per minute.

Arizona Copper Co., Morenci, Ariz. Stoping with Waugh air-feed drills, in rather hard porphyritic ore, requiring no timbering and worked by the block caving system. Eight-hour shifts. In a period of 21 working days (Jan., 1910), 260 holes were drilled; total depth, 1,433 ft., average depth, 5½ ft. Drilling time, 137 hours; delays, for picking down loose ground, etc., 31 hours. Average drilling speed, excluding delays, 10.46 ft. per hour, or 2.09 inches per minute.

Michigan Copper Mine, Rockland, Mich. Stoping in amygdaloidal brecciated vein matter, with Murphy air-feed drill, 628 feet of hole were drilled at the average rate of 42 ft. per shift of 9 hours, or 4.66 ft. per hour = 0.93 in. per min., including all delays. Two 6-foot holes were drilled on a test run at an average speed of 3 in. per minute.

Snowstorm Mine, Idaho. Stoping in well-mineralized quartz-

ose ore, with Waugh air-feed drills, 14 holes were drilled as follows:

TABLE XLVI.

Hole.	Depth, Inches.	Total Time, Minutes.	Net Drilling Time, Minutes.	Inches Drilled per Minute.	
				Total Time.	Net Time.
1	15	10	8	1.50	1.87
2	24	13	9	1.85	2.77
3	8	19	10	hole lost.	
4	25	15	8	1.55	3.12
5	26	12	5	2.16	5.60
6	31	19	12	1.82	2.58
7	34	20	14	1.70	4.42
8	44	11	7	4.	6.30
9	48	11	8	4.36	6.
10	53	33	21	1.60	2.52
11	48	17	10	2.82	4.80
12	51	40	15	1.28	3.40
13	46	20	12	2.30	3.82
14	49	18	11	2.72	4.45
Total	502	258	150	30.08	50.07
Averages	35.8			2.15	3.58

South Crofty Mine, Cornwall, England. The makers of the Climax Imperial hammer drill state that in Feb., 1909, one of their machines, column-mounted, made 34 holes in granite, averaging 37 inches deep, in 7 hours total time, or 3 inches per minute.

Village Deep and New Reitfontein Est. Mines, South Africa. In Dec., 1908, and Jan., 1909, 203 holes, aggregating 669 feet, were drilled by a Climax Imperial drill in 10½ shifts, or 63.7 ft. per shift; a general average of 1.58 inches per minute.

Barre Quarries, Vermont. The Sullivan Machinery Co. state that in Barre granite, their larger hand hammer drill, used in dimension stone work, makes a 6-inch hole in 1 to 1¼ minute.

Midlothian Colliery, near Richmond, Va. Sinking a rock slope in rather soft, coarse-grained sandstone, with three Hardsocg hand

hammer drills; 95 lbs. air pressure. Cross-section of slope, 7 ft. x 16 ft.; 27 four-foot holes = 108 linear feet per round; an average of about 18 rounds are made per month of 26 days, with three 8-hour shifts, the corresponding advance of the slope averaging 65 ft. Actual drilling time not recorded.

Esmeralda Mine, Silverton, Colo. Stoping with Waugh drill; holes 1 and 2 in medium hard andesite, hole 3 in hard quartzose ore and hole 4 in rather soft vein rock:

TABLE XLVII.

Hole.	Depth, Inches.	Total Time, Minutes.	Net Drilling Time, Minutes.	Inches Drilled per Minute.	
				Total Time.	Net Time.
1	35	13	10	2.70	3.50
2	58	27	21.5	2.15	2.70
3	49	23	18	2.13	5.16
4	62	14	12	4.43	5.16
Averages	51	19.2	15.4	2.85	3.52

Burra-Burra Mine, Tennessee Copper Co. Stoping with Murphy and Waugh drills, in medium hard pyritic ore. Average speed for 9 "uppers," ranging from 52 to 72 inches deep and averaging 63.5 inches, was 2 inches per minute, total time, including changing bits. Maximum speed, 3 inches; minimum speed 1.6 inches per minute. The usual duty per 9-hour shift is from 40 to 50 feet of hole, or 4.4 to 5.5 ft. per hour = 0.85 to 1.10 inches per minute, general average.

Los Angeles Aqueduct, Los Angeles, Cal. Tunneling in medium to hard, close-grained granite, favorable for drilling; Leyner drills Nos. 7 and 9, mounted on a horizontal bar; air pressure about 90 lbs. Test runs were made to compare the work of the two sizes of machine.*

* Abstracted from an article by J. B. Lippincott, *Engineering News*, April 22, 1909, p. 449.

Table XLVIII.

Hole.	Depth, Inches.	Total Time, Minutes.	Net Drilling Time, Minutes.	Inches Drilled per Minute.	
				Total Time.	Net Time.
No. 9 Drill. Weight of Hammer, 13½ Lbs. 600 Strokes per Minute.					
1	79	9.	7	8.77	11.30
2	87	36.25	16.75	2.40	5.20
3	96	29.25	19.25	3.28	4.98
Averages	87.3	24.83	14.33	4.82	7.16
No. 7 Drill. Weight of Hammer, 7 Lbs. 1,600 Strokes per Minute.					
1	83.5	7.75	6.50	6.08	12.80
2	83	19.50	16.00	4.25	5.19
3	84	19.50	16.00	4.30	5.25
Averages	83.5	15.58	12.83	4.88	7.75

The rock in which the first hole was drilled by each machine was much softer than that of the others.

Field of Work. It may hardly be questioned that, for such work as the driving of tunnels or cross-cuts of large section, or for underhand stoping in wide veins and in general wherever deep holes can be advantageously drilled and blasted, the majority of the hammer drills (omitting large machines of the Leyner type), cannot be expected to compete with reciprocating drills. But, in connection with these operations, hammer drills may often be made useful as auxiliaries, not only for block-holing but for "squaring up," after the main rounds have been fired; that is, dressing the walls, taking up the bottom when the deep holes fail to break clean, etc. For other mining work, also, as for example, drifting and stoping, either breast or overhand, the air-feed machines are useful; and, when mounted on bars or columns, will often give quite as good results as the 2 or 2¼ inch reciprocating drills, providing deep holes are not required.

The comparison is most favorable to the hammer drill in quarry work—both dimension stone and block-holing—and when stoping in thin veins, especially where the values occur chiefly in narrow pay-streaks. In the latter case, shallow holes, with correspondingly light charges, are desirable, to avoid scattering the rich ore and breaking it with the poor ore or wall-rock, which would have to be sorted out.

For shaft-sinking the case is not so clear. When the rock is solid with few fissures, slips, or bedding planes, reciprocating machines, capable of drilling 6, 8 or in large shafts even 10-foot holes, produce the best results. But, in shaly ground, or any rock that is full of slips and short fissures, deep holes often break imperfectly. It is here that the hammer drill can be used with advantage, because such ground is worked best by a relatively large number of shallow holes, and to drill them with reciprocating machines involves loss of time in shifting and setting up.

General Conclusions. The opportunity for substantial variations in the design of hammer drills is relatively limited, the largest differences being in the arrangement of the air ports and passages. Since both valve and rotating mechanism are usually omitted in the smaller sizes, most of these drills have but one moving part—the hammer—and their strong construction and simplicity adapt them well for the rough usage unavoidable in mine and quarry work. In the matter of repairs, therefore, they compare favorably with reciprocating drills.

Economy in the use of air is also a feature of the hammer drill. The smaller machines use say 25 to 30 cu. ft. of free air per minute; the larger, including the air-feed drills (but omitting the "Water" Leyner), from 35 to 55 cu. ft. per minute. Air pressures of at least 80 or 90 lbs. (sometimes 100 lbs.) are generally recommended; that is, the same pressures as are suitable for the large reciprocating machines. Some foreign machines, for example, the Gordon, which has done remarkably good work,* seem to be best adapted for pressures of not over 50 or 60 lbs.

* In the latter part of 1907, a series of tests was made on a number of small stoping drills, both reciprocating and hammer, at Johannesburg, South Africa.

The objection is sometimes made to hammer drills that, in drilling dry holes, they raise much dust. With the small hand machine, the operator must necessarily stand close to the mouth of the hole and, when the latter is cleaned by exhausting through a hollow bit, the dust is blown back directly in the operator's face. This is specially troublesome—and hurtful—in drilling either breast-holes or uppers (above the horizontal). It can hardly be denied that hand hammer machines do not appear to best advantage when drilling holes in such positions. The Murphy hand drill is provided with a ring of sponge, around the bit near the mouth of the hole. By keeping the sponge wet the dust difficulty is diminished, though not eliminated. Miners often neglect to care for such appliances. For work of the character mentioned, and for stoping in general, the air-feed machines, held by the operator or mounted on a bar, are to be recommended, together with the use of water injected through the hollow bit, or an external spray, as in the Climax drill, whenever the hole fails to clean itself.

Solid bits have only a limited application for hammer drills. They can be used for shallow holes in the softer rocks, by watering the hole and frequently spooning out the sludge; or in dry rock, provided the holes are inclined at a sufficient upward angle to permit the cuttings to run out by gravity. But they are best used for cutting hitches for mine timbers, block-holing, and quarry work.

An important feature of hammer drills with air-feed attachment is, that they can be set up or shifted in less than two minutes, under normal conditions; while the time required for shifting an

At the air pressure used (60 lbs. maximum), the Gordon drill stood first, drilling 36 ft. 9 ins. in 4 hrs., in granite; the next best record being 33 ft. 2 ins., by the Chersen, a reciprocating machine. But it is probable that, with a higher air pressure the results of the test would have been quite different, and more favorable for some of the other machines, designed for higher pressures. Weaknesses of construction were subsequently discovered in the Gordon drill, which is not now on the market. Full details of these tests, with tabulations of the work done by each machine, were published in the proceedings of the *Transvaal Institute of Mechanical Engineers*, January 11, 1908.

It may be added that since then the Climax hammer drill has done equally fast work.

equivalent 2 or 2¼-inch reciprocating machine, mounted on a column, is rarely less than 15 minutes and sometimes half an hour. The relative drilling capacity of the light hammer drills is thus increased. Still another saving in time results from the fact that the bit is loose in the chuck, so that for changing bits there are no chuck-bolts to be manipulated.

Hammer drills supply a substantial need in mining and quarrying, and in their proper field of work will unquestionably be more and more widely used.

CHAPTER XXII

COAL-CUTTING MACHINERY

Coal-cutting machines are extensively used to replace hand labor in the work of "under-cutting" the coal, preparatory to breaking it by blasting or wedging. The objects in view are: (1) To economize in the cost of mining; (2) to decrease the proportion of "fines" produced; (3) to increase the rate of production of coal from a given extent of mine workings. Coal cutters find their chief application in bituminous collieries, and under proper conditions it is unquestionable that by their use coal can be produced more cheaply than by manual labor alone. They are designed to groove or undercut the face or breast of coal, close to the floor and to a depth of several feet. The mass so undercut is subsequently broken down by a few comparatively light blasts. Fig. 163 shows a typical case of a reciprocating or pick machine, working in a thin vein.

Coal cutters may be divided into four classes:

1. Endless chain machines.
2. Rotary bar machines.
3. Disc or circular saw machines.
4. Reciprocating or pick machines.

The last named imitates in some respects the operation of a miner's pick. All of them may be driven by either electricity or compressed air, though electricity is now generally employed for cutters of the first three types. These will be only briefly described. Pick machines are almost invariably operated by compressed air, no entirely satisfactory electric driven pick having yet been placed on the market.

Endless Chain Cutters are built by several concerns, one example, the Jeffrey (type 16–D), being shown in Fig. 164. It has a

stationary bed-frame, consisting of two parallel steel channels, with the necessary cross-ties or braces. Within this frame is a T-shaped sliding frame, on the rear end of which is mounted the driving engine—a pair of small compressed air cylinders, say 5 in. diameter by 5½ in. stroke. The sliding frame carries an endless sprocket-chain, driven by a sprocket-wheel and gearing from the engine. Each alternate link of the chain is provided with a socket, in which is set a cutting tooth or bit. The head of the sliding frame is

FIG. 163.—Sullivan Coal Pick, Working in a Thin Vein.

composed of two parallel steel plates, as shown, between which, at the forward corners, are placed the idler sprockets carrying the chain. Mounted on each end of the frame are screw-jacks, for bracing the machine firmly in position while at work. The front jack is set against the face of coal, the other against a post or prop.

The cutting bits, held in their sockets by set screws, are straight-edged and point forward in the direction of motion of the chain. They are so shaped and staggered with respect to one another, as to "cover" the chain and cutter head, and make a groove in the coal about 4 inches in height, or sufficient to permit

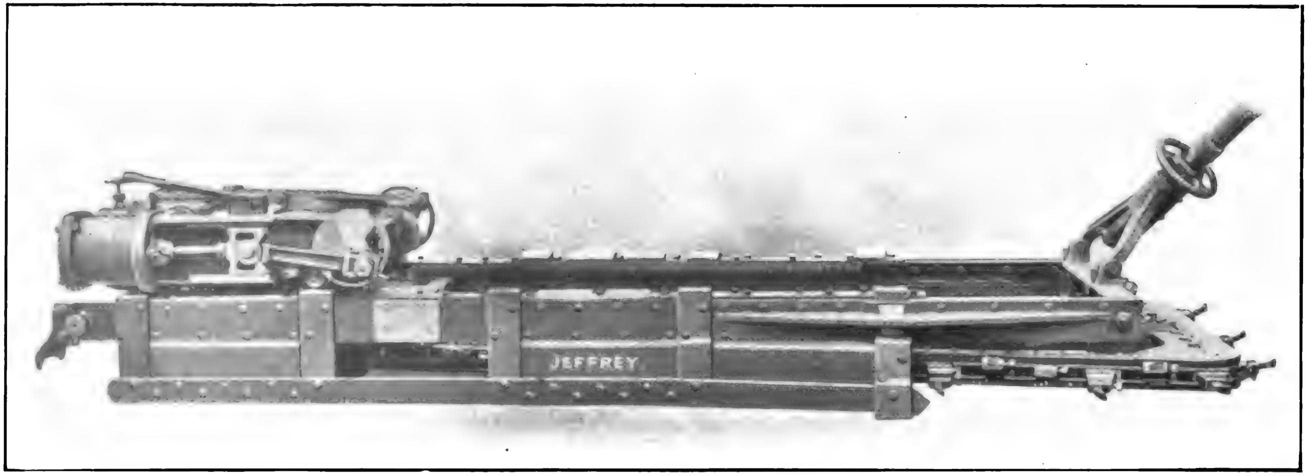


Fig. 164.—Jeffrey Chain Machine.

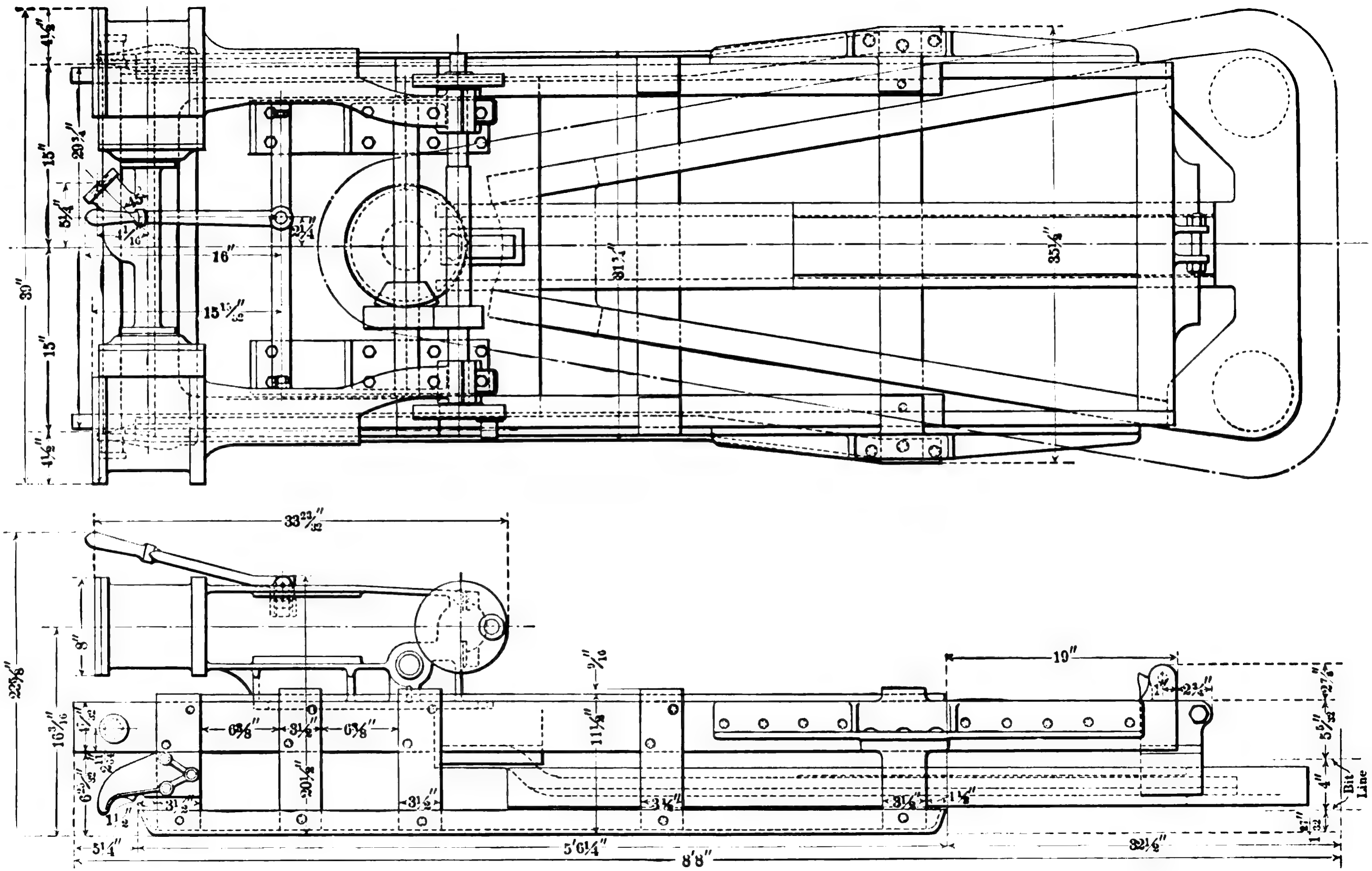


Fig. 165.—Jeffrey Chain Machine. Plan and Elevation.

the cutter head to enter freely as the work advances. The sliding frame is fed forward automatically as explained below.

Fig. 165 shows the general plan and elevation of this machine, with some of the principal dimensions, and Fig. 166 is an enlarged plan and elevation of the compressed air engines and their accessory mechanism. Referring to Fig. 166 the engine crank-shaft *a* drives the countershaft *b*, through the gear wheels *c* and *d*. Keyed on *b* are: (1) The bevel gear *e*, engaging with bevel gear *f*, on the short vertical shaft *g*, which carries on its lower end the sprocket *h*, for driving the cutter chain; (2) the worm *i*, which, through the shaft *j* (and by another worm, as shown in the elevation), drives the gear wheel *k*, mounted loosely on the transverse shaft *l*. At each end of *l* are the small pinions *m*, *m*, engaging with long feed racks, bolted to the side channels of the stationary or main frame; (3) the small bevel gears *n* and *o*, which, through the shaft *p*, drive the worm *q*, engaging with the gear *r*, also mounted loosely on the transverse shaft *l*.

The effect of these two trains of gearing is to cause *k* and *r* to rotate in opposite directions, *r* having the more rapid movement. By means of the sliding clutch *s* on the shaft *l*, either *k* or *r* can be thrown in gear. When *k* is in gear the sliding frame carrying the driving mechanism, with the cutter head and chain, is fed slowly forward as the undercut advances. When *r* is thrown in gear, after the cut is finished, the feed is reversed and the sliding frame and cutter head are rapidly run back, preparatory to shifting the machine for the next cut.

The depth of undercut made by chain machines ranges from 4 to 7 feet; width of groove, from 39 to 44 inches. By shifting the machine, successive cuts are made side by side, over the required length of face. In regular operation, from 100 to 150 square yards can be undercut in 10 hours, depending on the character and hardness of the coal and upon whether the work is in "rooms" or "long-wall" mining. Two men are required: a machine runner, and a helper to remove the débris from the undercut and assist in handling the machine. Tests show that the power required varies from 8 to 14 horse-power, an average of 12 horse-

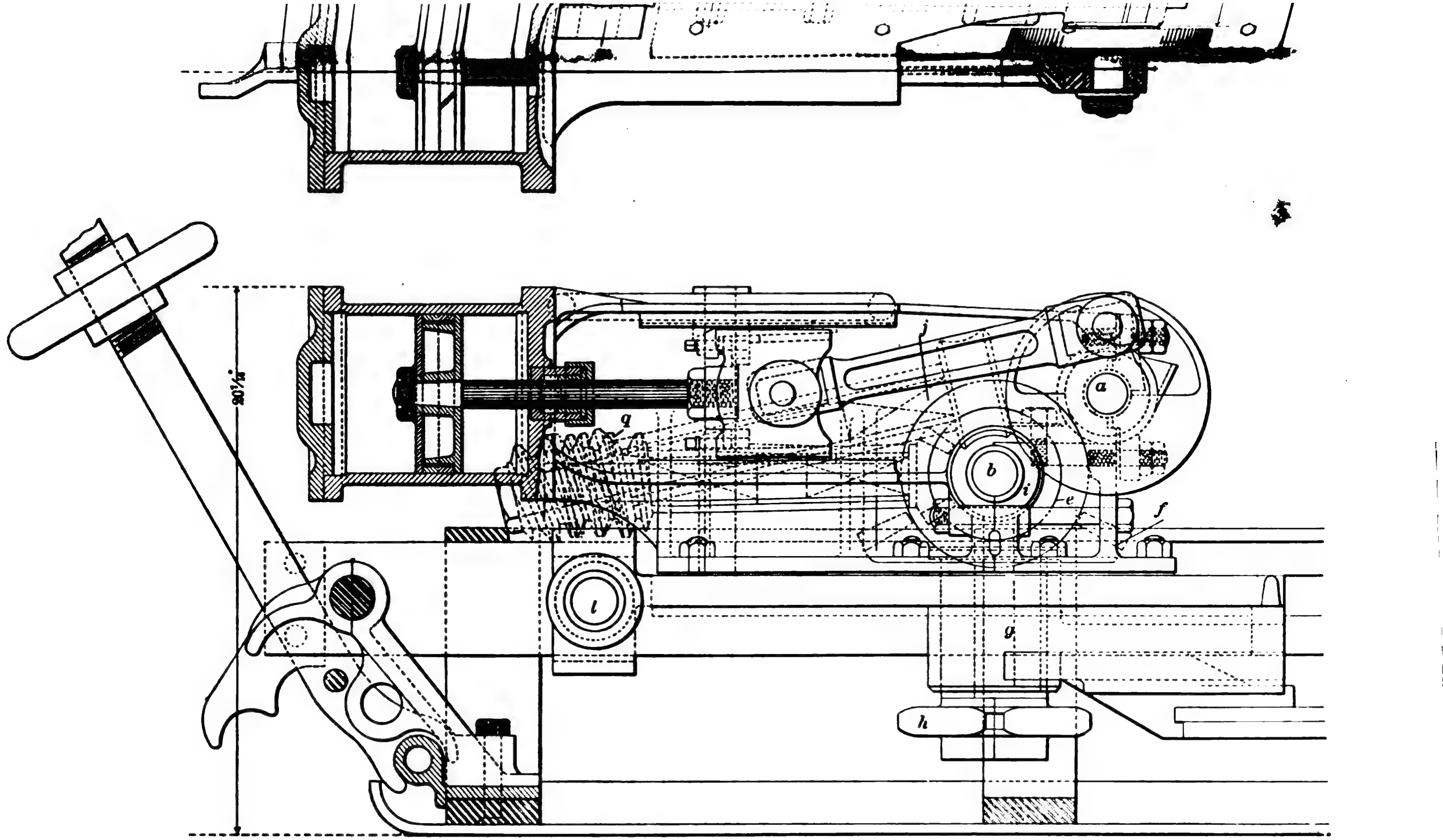
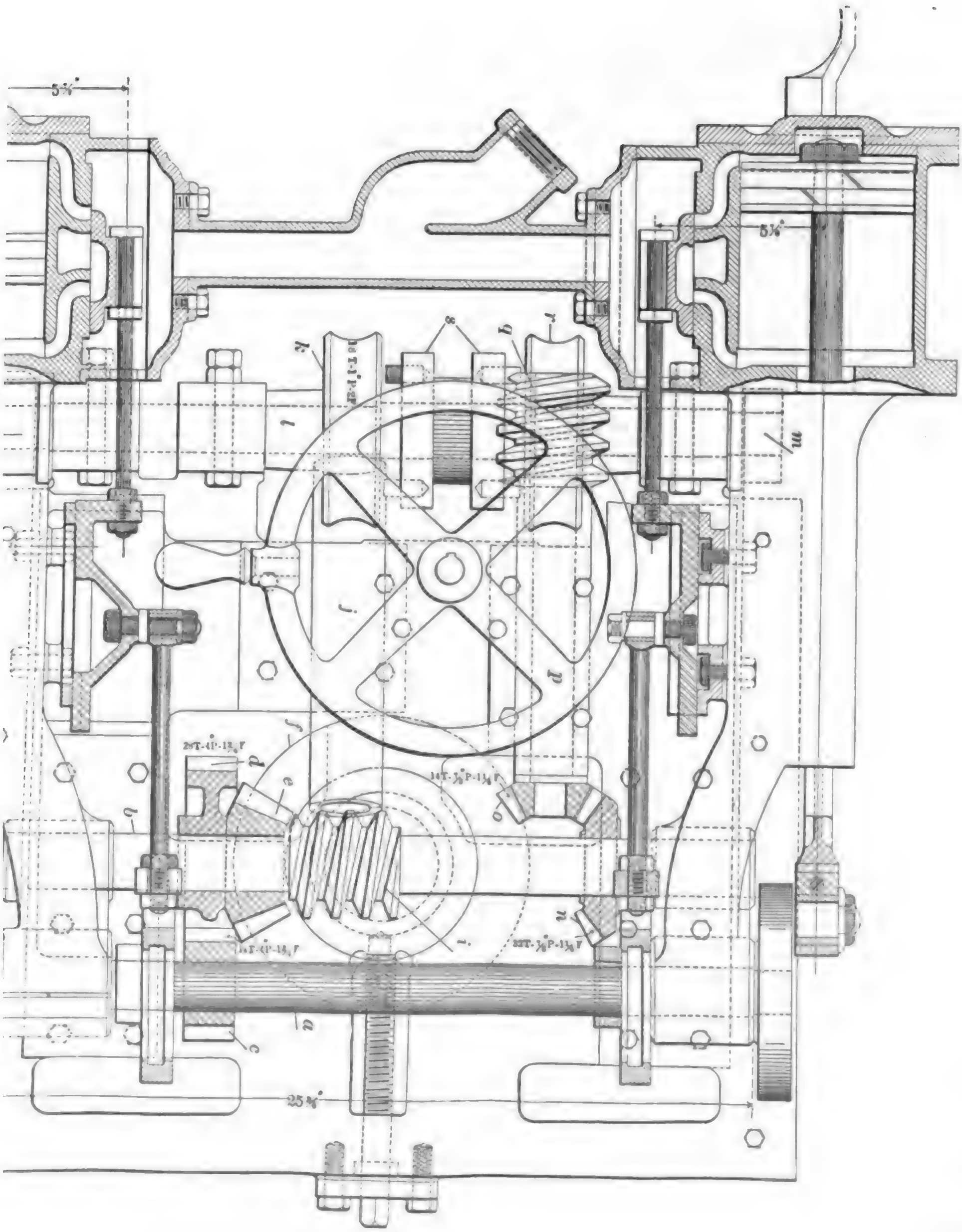


FIG. 166.—Jeffrey Chain Coal Cutter. Plan and Elevation of Compressed Air Driving Engine.



power being obtained from numerous tests in coals of varying hardness and quality.

Chain cutters of a number of different makes are in use, operated by either electricity or compressed air. Among them are those of the Jeffrey Manufacturing Co., Sullivan Machinery Co., Link Belt Machinery Co., and the General Electric Co. (Sperry machine). For long-wall mining some of them are self-propelling and operate continuously along a face of coal of any desired extent. A post is set at one end of the face and to it is attached a chain leading to the cutter, the chain being wound in by a small drum, geared to the driving engine.

Rotary Bar Cutters are not so widely used as the chain machines. They have a bed-frame similar to that of the chain cutter and the inner sliding frame is fed forward in the same way. The cutter head carries a horizontal shaft or bar, in which is set a series of bits. This cutter bar is rotated by a sprocket-chain driven from the air engine (or electric motor) in the rear. The bar is from 3 to 3½ feet long, the width of undercut being slightly greater, owing to the projection of the end bits; speed of rotation of the bar, about 200 revolutions per minute. Under favorable conditions the time required to make a cut 5 to 5½ feet deep ranges from 6 to 7 minutes. In room work, from 70 to 100 square yards can be undercut in 10 hours; more in long-wall work. The power required usually ranges from 14 to 16 horse-power.

Disc or Circular Saw Cutters are made and used almost exclusively for "long-wall" mining. In the past they have been employed more commonly in Europe than in this country. Their general construction will be understood by reference to the accompanying illustrations of the Jeffrey (American) machine, style 22-C. Figs. 167 and 168 are perspective views of opposite sides and Fig. 169 a plan and side elevation, on which only a part of the periphery of the cutter wheel is indicated.

The driving machinery is compactly arranged on a bed-frame, mounted on wheels, to run on a temporary track laid along the face of coal. (The machine shown in the cut requires but a single line of rails.) A cutter wheel, from 3 to 4½ ft. in diameter,

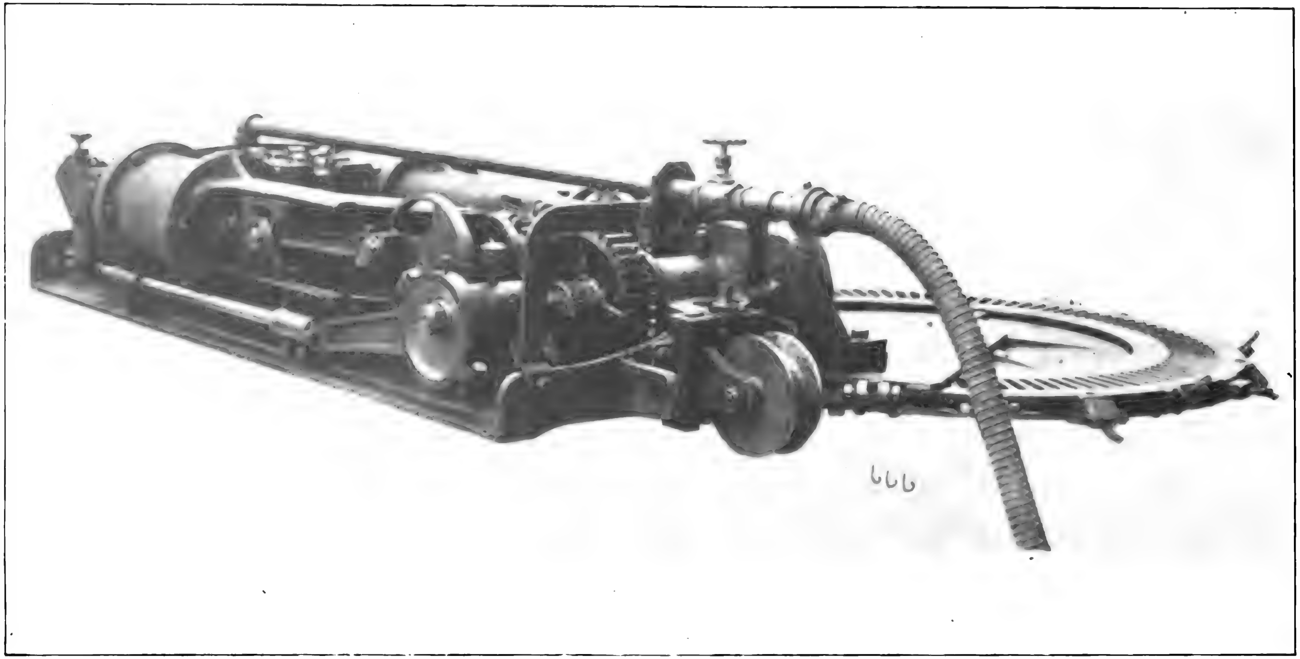


FIG. 167.—Jeffrey Disc Coal Cutter, Style 22—C.

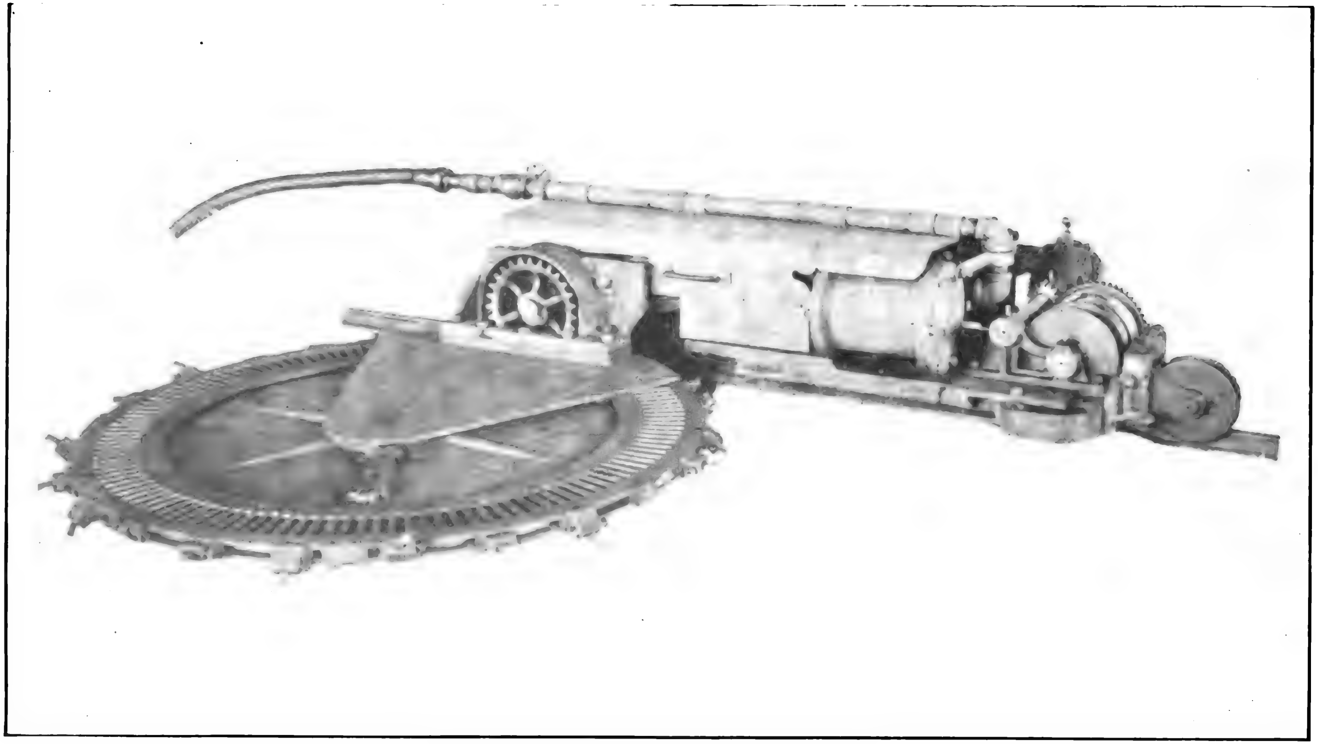


FIG. 168.—Jeffrey Disc Coal Cutter, Style 22—C.

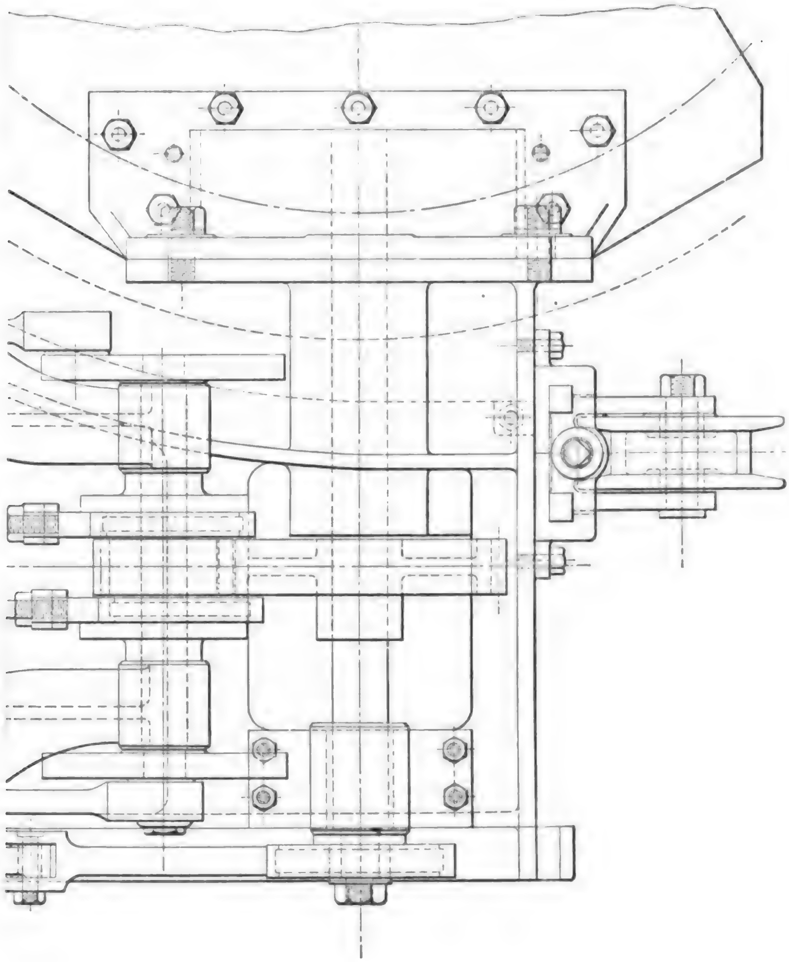
is supported by a heavy bracket on one side of the frame, and projects a distance nearly equal to its own diameter. On the periphery of the wheel is set a series of staggered bits, cutting a groove of sufficient height or width, to admit the disc freely. The cutter wheel is driven by double reduction gearing, from a pair of compressed air cylinders, with quartering cranks. The speed of the wheel ranges usually from 15 to 30 revolutions per minute, depending on the character of the coal. When in operation the whole machine is automatically pulled along the face of coal by a small wire rope, made fast to a post at the end of the face, and wound in by a drum geared to the driving engine.

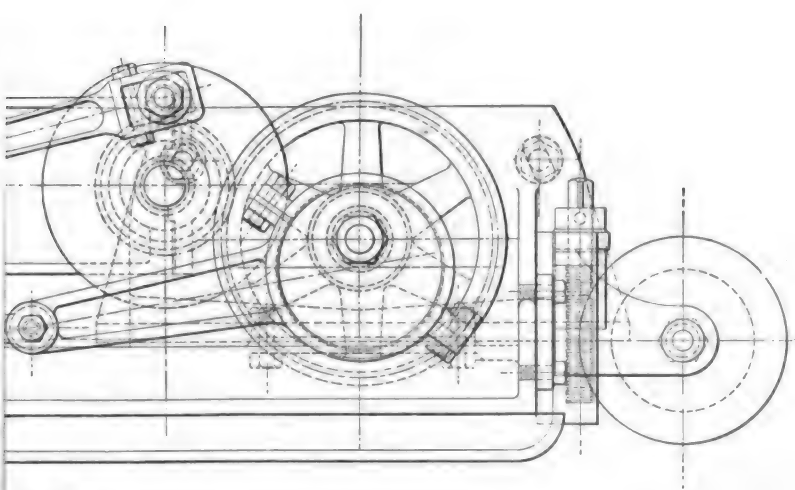
The disc cutter is a useful machine for long-wall work, specially in thin or steeply pitching veins, where it would be difficult or impossible to operate cutters of the other types. It works back and forth along the face, almost as well up the dip as down. The linear speed of feed is from, say, 8 to 25 inches per minute. These machines will undercut from 90 to 100 square yards per 10 hours, under average conditions.

Among the English machines of this class may be mentioned the Gillott and Copley, Yorkshire, Rigg and Meiklejohn and the Winstanley. Though they vary in details, their general construction and operation are essentially as described above.

Reciprocating or Pick Machines.—At the present time these constitute the most important of the four classes of coal cutters. In their general construction they possess many points in common with the reciprocating rock drills; which, in fact, have furnished the basis of the design of several well-known makes of pick machine. All work without rotation of the piston and bit, since in undercutting there is no question of preventing rifling of the hole, as in rock-drilling. Variations in the valve-motion are noticeable, some of the designs being entirely different from the spool- and tappet-valve motions.

The first pick machine was invented in 1858, by E. Simpkins, of Allegheny City, Pa., but it was of crude design, imitating the movements of the miner's pick. Next in order of seniority is probably the Harrison machine, invented in 1877. This was fol-





-C. Plan

lowed in 1881 by the Yoch coal pick. From time to time both of these machines have been altered and improved in many of their features. Of later date are the Sergeant and the Sullivan picks. All of the above are operated by compressed air.*

The general lines of this class of coal cutter, together with the mode of operation, are shown in Figs. 163, 170 and 171. The machine is mounted on a pair of wheels and when at work is placed on a wooden platform, about 3 ft. wide by 8 ft. long, which slopes towards the face of coal, at an angle of, say, 5 degrees. By this means, the recoil of the blows is nearly neutralized by gravity and the machine is kept up to its work. The operator chocks the wheels with wooden blocks, sometimes strapped to his boots, and directs the blows by swinging the machine from side to side, with the supporting wheels as a fulcrum. As shown by the several cuts, the front cylinder head and piston rod are very long, to give the machine a sufficient reach. A horizontal width of 4 or 5 ft. of undercut is thus readily commanded. The depth of cut is rarely greater than 5 feet. A helper clears away the débris with a light, long-handle shovel, and assists in moving and setting up the machine.

Most pick machines run at speeds of 200 to 250 strokes per minute. The lower speed machines probably have some advantage, because, as each individual blow is directed by the operator, he can increase the efficiency of the work if he has time between strokes to point the pick in such a manner that it will do most execution. In coal of average quality, an undercut of say 4 ft. by 4 ft. in horizontal area can be made in 16 to 18 minutes. The platform can be shifted sidewise to the next position and the bit changed, if necessary, in 8 to 10 minutes. The height of undercut is 12 to 14 inches at the face, tapering to 3 in. or 3½ in. at the bottom. On completing the cut, the coal is shot down by light blasts; or, in gassy mines, is sometimes broken by wedging. Under favorable conditions, good operators can undercut, per shift, from 75 to 85 linear feet of face, to a depth of 4 to 4½ feet;

* It may be added that an electric-driven coal pick, the Thomson-Houston solenoid, has been put on the market.

FIG. 170.—Harrison Pick Machine at Work.

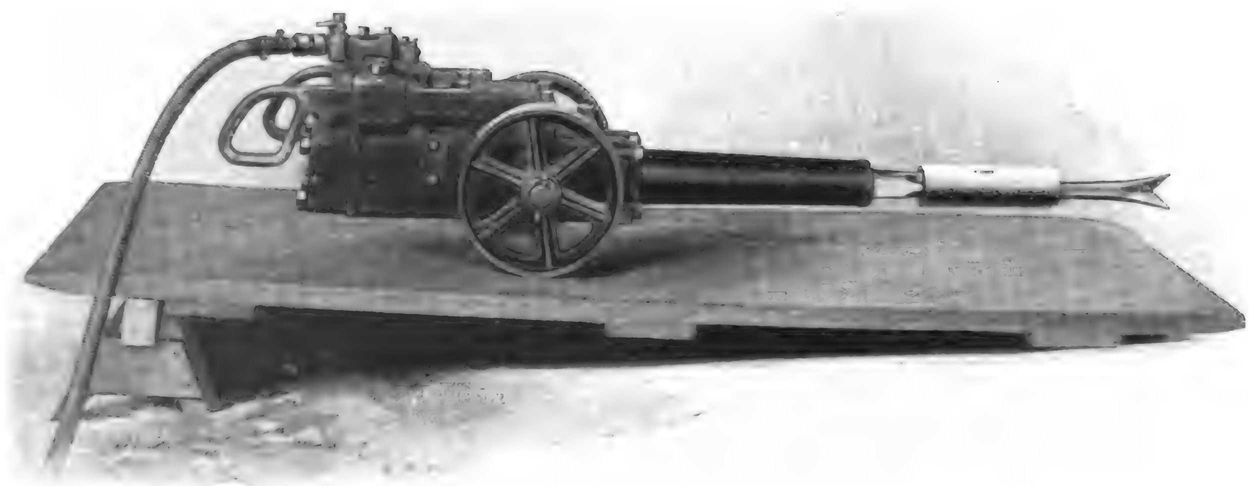


FIG. 171.—Sullivan Pick Machine Mounted for Work.

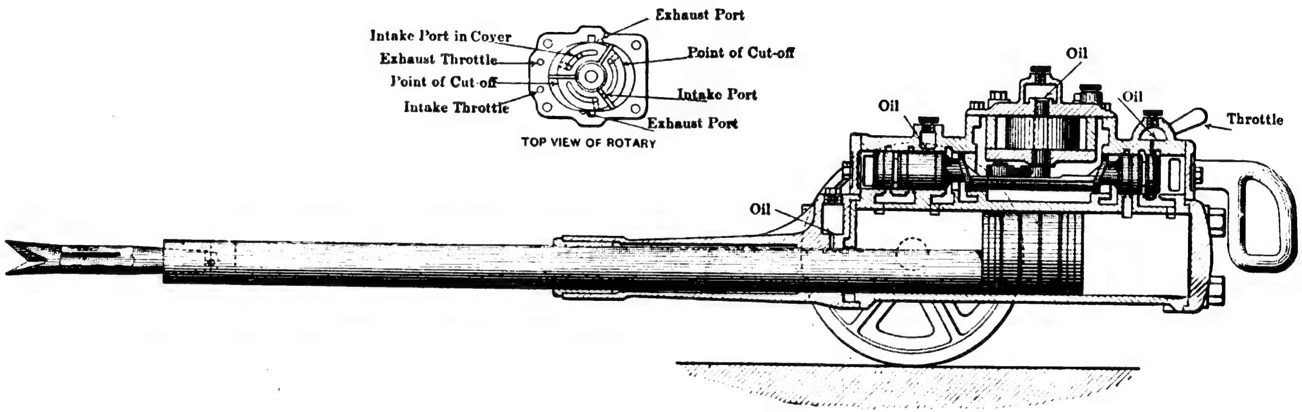


FIG. 172.—Harrison Pick Machine.

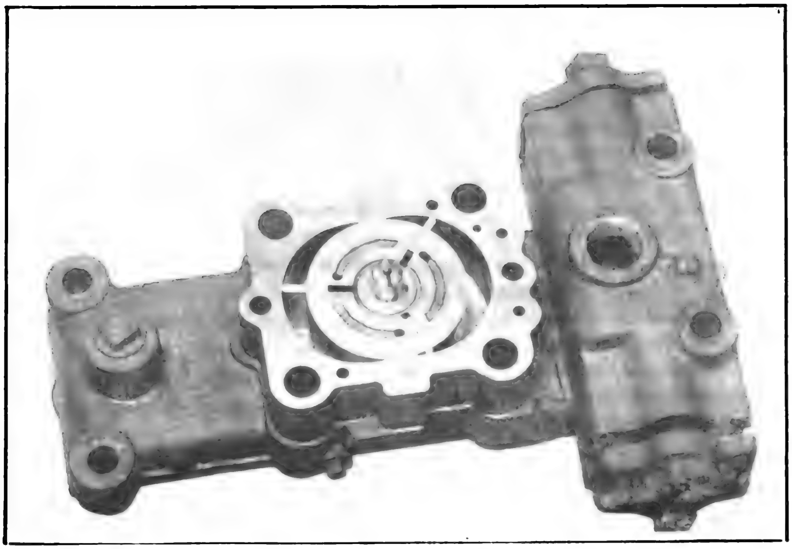


FIG. 173.—Rotary Engine for Operating Valve of Harrison Coal Pick. Top View.

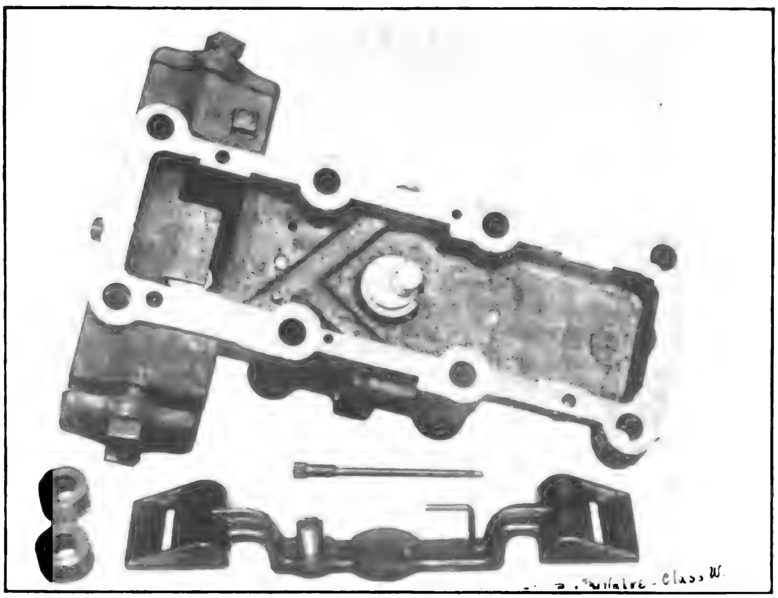


FIG. 174.—Rotary Engine for Operating Valve of Harrison Coal Pick. Bottom View.

in competition trials much faster work is often done. Fair, average work would be from 60 to 65 ft. of undercut, 4 ft. deep, per shift.

Harrison Pick Machine.—Fig. 172 shows a longitudinal section of models "P G" and "P W" of this machine. The valve is a long, double spool, actuated through a crank and connecting rod by a small horizontal rotary engine set above the middle of the valve chest. A separate plan of the rotary, with its ports, is also shown in the cut; the main cylinder has double ports at each end, to

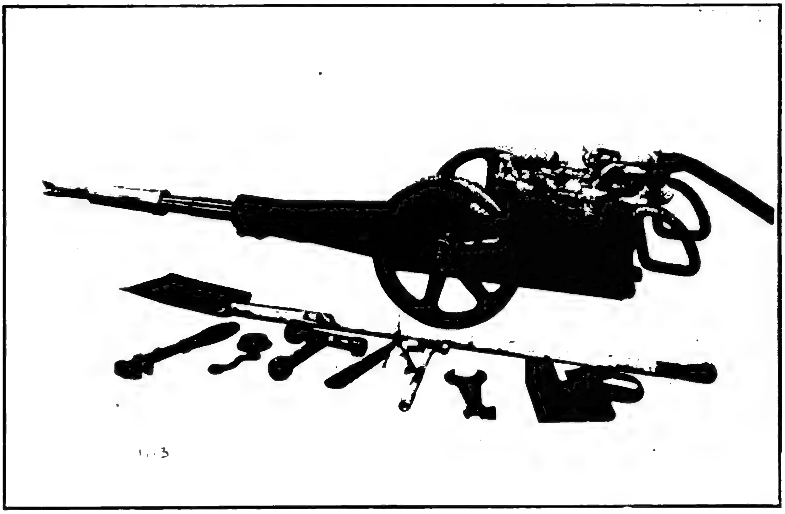


FIG. 175.—Ingersoll-Rand Coal Pick.

obtain cushioning of the stroke, and to enable the machine to run with a short stroke when desired. Figs. 173 and 174 are perspectives respectively of top and bottom of the valve motor for models W of the Harrison coal pick, the latter showing also the long slide valve and its connecting rod.

These machines are made in heavy and light patterns, the former weighing about 700 lbs., the latter 500 lbs. The heavier pick will cut to a depth of 5 ft. and is specially adapted for "shearing"; that is, making vertical cut or grooves, on one or both sides, in driving entries for developing a coal seam, or for haulage- and

air-ways. For this purpose, supporting wheels of 34 ins. or 40 ins. diameter are provided, in order to raise the machine high enough to give the requisite reach. Smaller wheels are used for ordinary undercutting.

"New Ingersoll" Pick Machine (Fig. 175).—The valve-motion of this machine is developed from the spool-valve rock drills of the Ingersoll-Rand Co., with the important difference, however, that the air ports are controlled by a combination of slide- and spool- or plunger-valves. Fig. 176 is a longitudinal section and Fig. 177 a diagrammatic sketch of the valves, ports and chest, developed in one plane to show the relations between the several ports and the operation of the valve motion.

The main ports S, S^1 of the cylinder O are controlled by the slide-valve G, on the back of which is a lug H, engaging with the double spool-valve F. Air is admitted alternately to the opposite ends of F through the auxiliary ports J, J^1 which in turn are controlled by the auxiliary slide-valve K and its actuating spool-valve F^1. This valve is

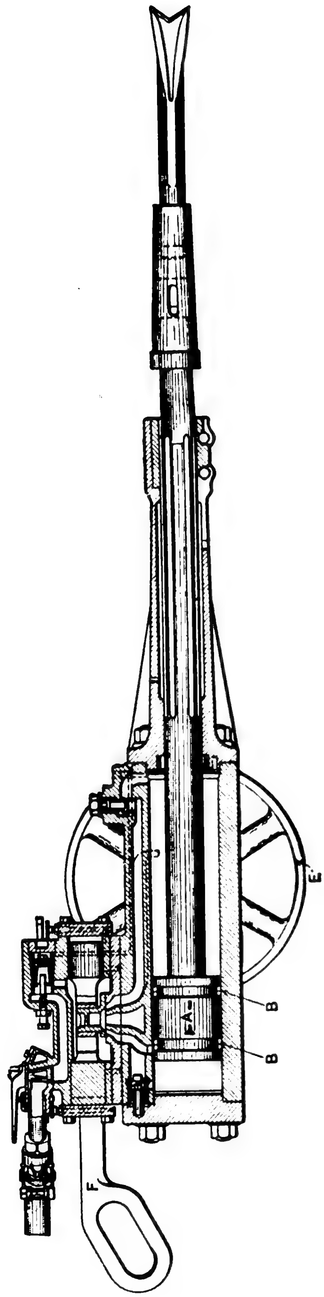


FIG. 176.—Ingersoll-Rand Coal Pick. Longitudinal Section.

thrown by air passing through the small ports N, N^1, N^2 and N^3, which, by a cross-over arrangement as shown, connect on either side with the main ports S, S^1. Thus the rear end of the chest of spool-valve F^1 is connected with main forward port S and the forward end of F^1 with the rear main port S^1; so that, when air is admitted to the main cylinder O through the rear

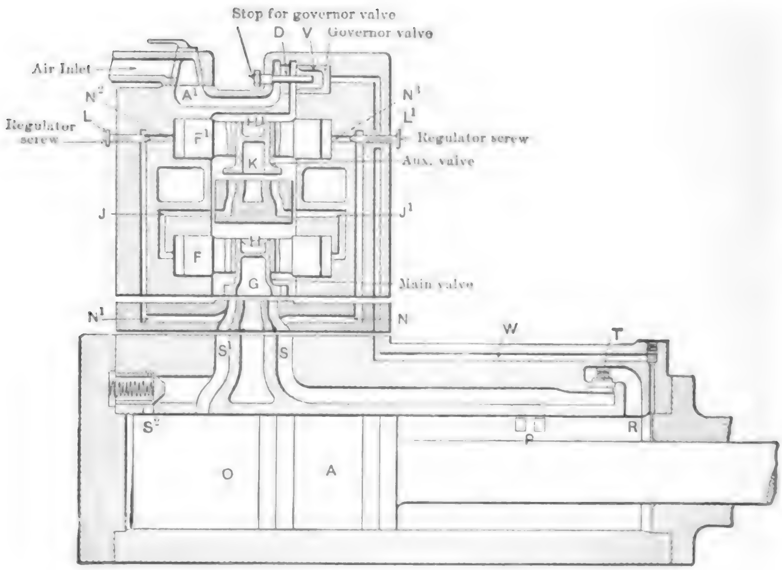


FIG. 177.—Ingersoll-Rand Coal Pick. Diagram of Valves and Ports.

port S^1, a small portion of it passes through N, N^3 and drives back the valves F^1 and K. This movement admits air from F^1 through port J^1 to spool-valve F and throws back the main slide valve G; thus opening main port S to live air and S^1 to the exhaust. Some of the air passes from S through the rear auxiliary port N^1, N^2 to the valves F^1 and K, by which first K and then G are thrown forward, reversing the main ports and completing the cycle of movements. As the main port S^1 is of larger area than the port S, the forward stroke of the machine is more powerful than the back stroke.

In addition to the above parts there are two small regulating screws L and L^1, by which the operator can regulate the speed of the auxiliary valves F^1 and K and hence of F and G. But, while the effect of the regulating screws is to govern the speed of stroke, they do not greatly influence the force of the blow. The forward stroke is cushioned, as follows: When the piston passes the double port P the exhaust ceases, since at this point the bit should strike its blow. Hence, if the pick misses the coal or should momentarily be swung out of contact with it, the piston on advancing beyond P compresses the air in the forward end of the cylinder to a pressure higher than that in the valve chest A^1, which is connected with the cylinder by the small port W. This high pressure air forces back the governor valve V, wholly or partly cutting off the inlet air on its way to valve K, so that the machine will run at reduced speed and force until the bit again strikes the coal before the piston has covered the port P. The regular exhaust then takes place, relieving the pressure transmitted through W to the governor valve, which opens automatically and the machine at once resumes regular operation. By means of the stop D the throw of the governor valve may be adjusted as desired.

The air on which the piston cushions is confined by the spring check-valve, T, in the port S; and the back stroke is begun by the elasticity of the cushioned air. No live air can enter this end of the cylinder until the piston has advanced far enough on the back stroke to reduce the pressure in the cylinder below that of the live air in port S, plus the resistance of the check valve spring. A smaller degree of cushioning is similarly produced on the back stroke by the arrangement of the port S^1 and valve S^2.

The Ingersoll pick is made in three sizes of cylinder, viz., $4\frac{1}{2}$ in., 5 in. and 6 in. diameter, but there are five models, varying in weight, power and length, according to the depth of undercut desired. The maximum depth of cut ranges from 4 to 6 feet and the gross weights of machine, from 550 to 950 lbs. Standard wheels for mounting the pick are 14 in. and 17 in. diameter. Larger wheels may be used for shearing.

Sullivan Pick.—Fig. 163 is a general view of the machine. It is

made in three sizes, with cylinder bores of $4\frac{1}{2}$ in., $4\frac{3}{4}$ in. and $5\frac{1}{8}$ in., which undercut to maximum depths of from $4\frac{1}{2}$ to 6 feet. The weights range from 650 to 850 lbs. Air consumption of the $4\frac{1}{2}$-inch machine is about 115 cu. ft. of free air per minute; of the $5\frac{1}{8}$-inch machine, 135 cu. ft.

A longitudinal section of one of the larger sizes is given in Fig. 178, accompanied by a descriptive list of the principal parts. It will be seen that in some of its features this machine differs greatly from those previously described. The spool- or piston-valve (123), by its reciprocations, throws the long flat valve (126), which in turn controls the main ports (148 and 149) and the secondary ports between them. These secondary ports serve to produce cushioning at each end of the stroke, in a manner similar to that of the Ingersoll-Rand machine. A check-valve (130) is inserted in the forward main port (149), and, when the pick does not strike the coal, the piston runs forward far enough to form an air cushion in the front end of the cylinder. This closes the check-valve, and prevents immediate admission of live air on the return stroke, the first part of which is made by the cushion air. Injury to the cylinder head is thus prevented, as well as annoying shocks to the machine runner.

LIST OF PRINCIPAL PARTS (Fig. 178).

100. Piston (bare).
104. Rifle Nut.
105. Rifle bar, with gear.
106. Seat for 109.
107. Spring pointer for 108.
108. Stem for adjusting 106
109. Reverse valve.
110. Valve plate.
111. Cover for 110.
113. Spring for 115.
114. Regulating valve.
115. Index lever for 114.
116. Head (bare) for 127.
117. Packing leather (large) for 123.
118. Ring for 117.
120. Binding screw for 118 and 122.
121. Ring for 122.
122. Packing leather (small) for 123.
123. Valve (piston).
124. Buffer for 123.
126. Valve (flat).
130. Check-valve.
134. Plug for oil hole.
135. Pick.
136. Chuck.
137. Head (front) for 142.
139. Bushing in 137.
140. Packing leather for 100.
141. Collar for 140.
142. Cylinder (bare).
144. Trunnion for wheel.
147. Drift-Key for backing out pick.

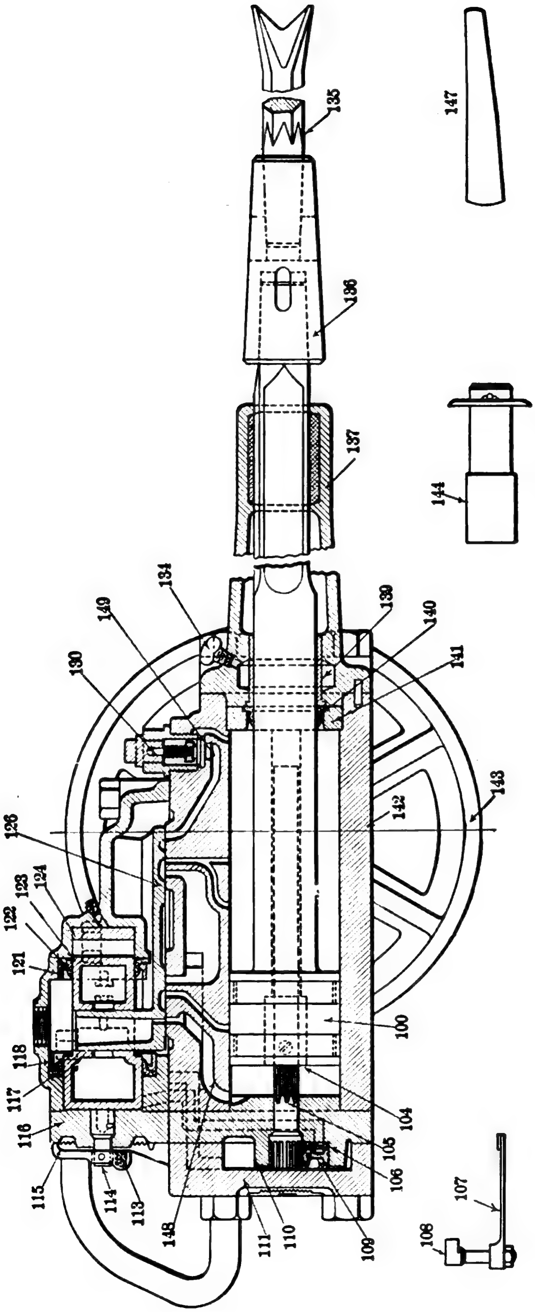


FIG. 178.—Sullivan Coal Pick. Longitudinal Section.

In the hollow piston is set a rifled-nut (104), with which engages the rifle-bar (105). As the forward end of the piston rod has flattened sides, making it nearly square in section, it cannot rotate like that of a machine-drill. Hence, at each stroke the rifle-bar is caused to rotate. The small gear cut on the back end of the rifle-bar causes the rotation of the reverse valve (106 and 109). This valve in turn controls the movements of the spool-valve (123),

FIG. 179.—Ingersoll-Rand "Radialaxe" Coal Cutter.

through the ports shown by dotted lines in the rear end of the cylinder. By means of a regulating valve (114), on the back of the main valve chest, the operator adjusts the speed of stroke, as the conditions under which the machine is working may require. The Sullivan pick runs at a moderate speed, which has the advantage of enabling the operator with more certainty to direct each blow where it will do the most efficient work.

Ingersoll-Rand " Radialaxe " Coal Cutter.—This machine is intended for shearing in entry work, as well as for undercutting.

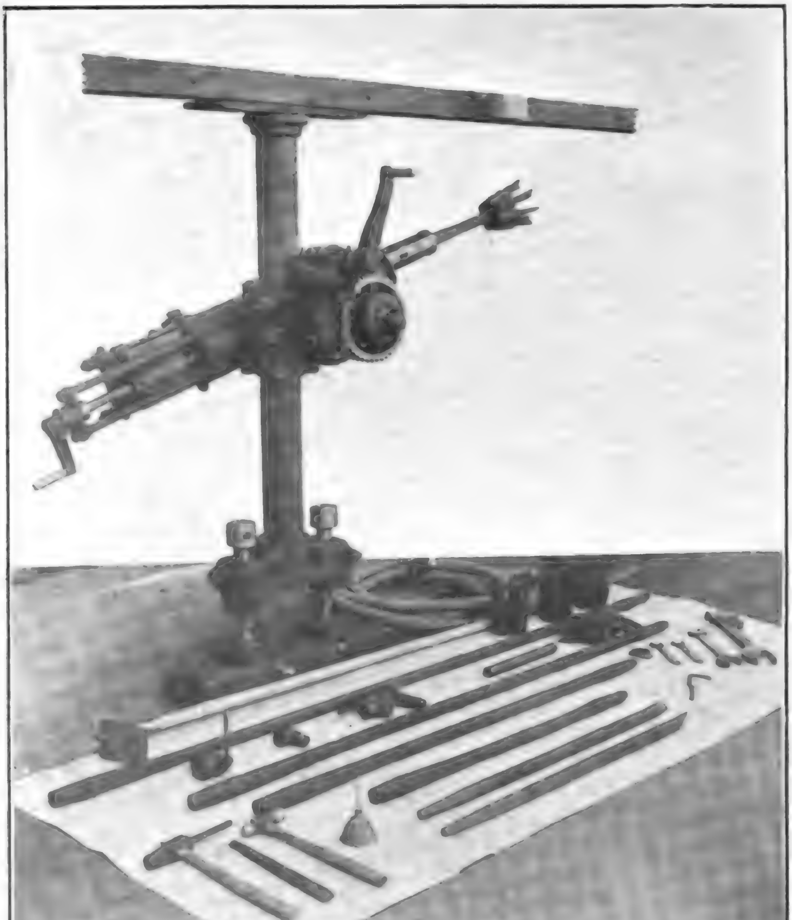


FIG. 180.—Ingersoll-Rand " Radialaxe " Coal Cutter.

As originally designed it consists essentially of a long-stroke rock drill, mounted on a column, and provided with a worm and worm-wheel sector. A hand wheel on the worm spindle enables the operator to swing the entire machine while at work, in either a horizontal or vertical arc. Long-shank bits are used, to give the machine the necessary reach.

The latest form of the "Radialaxe" as shown in Figs. 179 and 180 is an adaptation of the Temple-Ingersoll Air-Electric drill. In this design the gearing for swinging the machine is changed in some unimportant details. The bit, or group of bits, as shown, is set in a rosette socket, held by friction only on the tapering end of the drill shank, which is similarly inserted in the deep socket of a long chuck. The individual bits are thus readily removed for sharpening and replacement when broken.

Pneumelectric Coal Puncher.—Amongst the coal picks this machine occupies a class by itself. As its name implies, it combines the use of compressed air and electricity in a single piece of mechanism, consisting of a small electric motor, which drives a pair of independent pistons in a cylinder. This design may have been suggested by the Temple rock drill. Figs. 181 and 182 show the general construction. The casing T contains the motor, with its armature shaft A in a vertical position. To change its rotary motion to the rectilinear motion required for the piston, the following device is employed. The driving pinion B engages with the large horizontal gear-wheel C (Fig. 182), which has a solid web, carrying a stud D (Figs. 181 and 183). Mounted on D is a small gear E, and a crank with crank-pin G. Within the main gear C, and attached rigidly to it, is another gear F, with internal teeth, which engages with and drives the gear or crank-pinion E; F having 66 and E 33 teeth. Referring to the two diagrams in Fig. 183, it will be seen that the gears are so proportioned that the stud D revolves in a circle concentric with F and of one-half its pitch diameter. The gear E, which revolves freely on D, is carried with D, and causes the crankpin G and the cross-head H to reciprocate between the guides. To the cross-head is attached the piston rod I, with its piston J (Fig. 181).

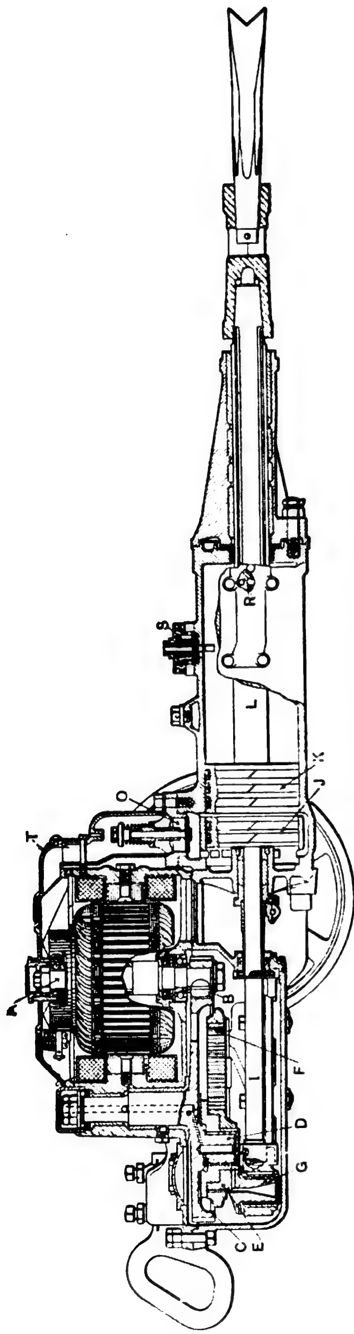


FIG. 181.—Pneumelectric Coal Puncher.

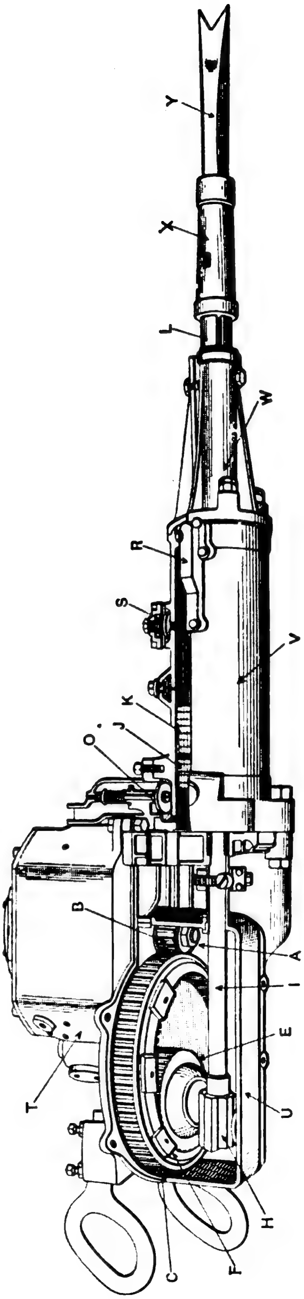


FIG. 182.—Pneumelectric Coal Puncher.

The cylinder *V* contains two pistons, entirely unconnected with each other: the rear or driving piston *J*, already noted, and the forward piston *K*, with its rod *L* and chuck *X*, for holding the bit *Y*. In starting the machine on its first forward stroke, piston *J* simply pushes *K* forward, air meantime entering the cylinder behind the piston, through the valve *O*. On the back stroke of *J*, the air in the rear of the cylinder is compressed, occupying the clearance space and the passage between the cylinder and the valve

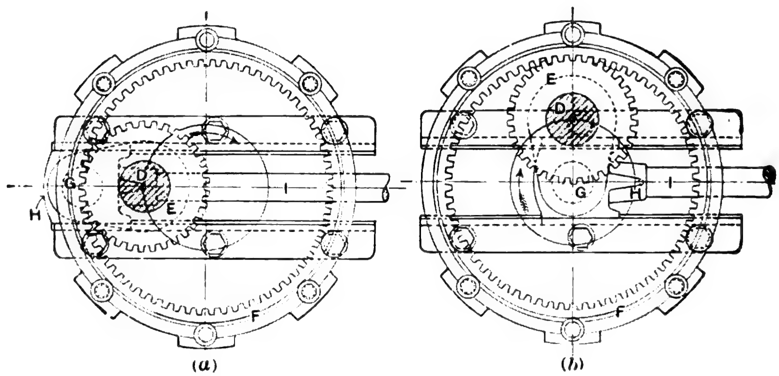


Fig. 183.—Pneumelectric Coal Puncher. Diagram of Gearing.

O. At the same time the air between the pistons is rarefied, thus causing *K* also to make its return stroke by suction, while air enters freely through a port at *R*. At the end of the back stroke, the air passages below the valve *O*, referred to above, lie between the pistons, whereby the charge of compressed air enters the cylinder and drives the piston *K* forward on its first regular stroke. The stroke is cushioned on air confined in the end of the cylinder, after *K* passes the port at *R*. Piston *K*, having completed its forward stroke, is followed by piston *J*, the air between them being discharged into the atmosphere through the valve *S*. The return stroke is then made by both pistons, as at first.

The diameter of the cylinder is 6¼ inches, and the clearance spaces at the rear end are proportioned to produce a final or working pressure of 95 to 100 lbs. The motor is designed to run at

three speeds, under the control of the operator, giving to the pick 140, 160 or 180 strokes per minute.* It is stated that 7 H.P. are required to run the machine, which is in successful operation at a number of collieries. The behavior of the motor, while at work, is shown by the following records, taken after two hours' operation.

Test.	Input.		Temperatures, Degrees Centigrade.				
	Amps.	Volts.	Room.	Armature Winding.	Commutator.	Brush.	Field Coils.
1	26	220	17	79	75	77	70
2	24	220	24	73	64	not recorded.	

The temperatures include room temperatures prevailing during the tests. It may be noted, respecting this machine, that, as the compressed air is released at the same point of every stroke, the

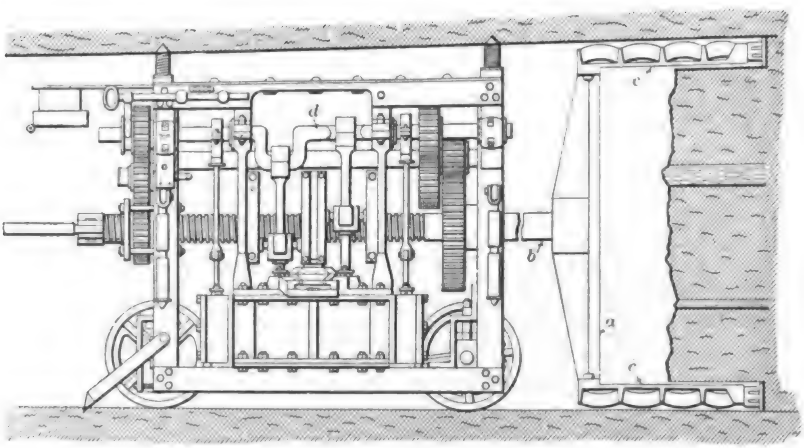


Fig. 184.—Stanley Heading Machine for Collieries.

power required of the motor is constant; in other words, the motor is not subject to overloads.

The Stanley Header, originally brought out in England, is intended for development work in collieries, driving circular head-

* This description is abstracted in the main from an article by Timothy W. Sprague, in *Mines and Minerals*, April, 1908, p. 427.

ings, for entries, airways, etc. In its earlier form (Fig. 184) a heavy cross-head, *a*, mounted on a central screw shaft, *b*, and carrying at each end a horizontal arm, *c*, cuts an annular groove from 3 to 4 inches wide. A central core of coal is thus left, which either breaks up and is shovelled back as the work advances, or is blasted or wedged down from time to time, if necessary. The operating mechanism consists of a pair of compressed air cylinders, about 9 ins. by 9 ins., from the crank-shaft, *d*, of which the screw-shaft is driven by gearing. Differential gearing is used to produce the feed, as shown. The whole is carried in a frame on wheels, held firmly when in operation by jack-screws set against the roof. The machine is narrow enough to permit a man to pass alongside to the front, to throw back the broken coal and keep the cutter head free.

Several modifications of the Stanley Header have been introduced in this country and abroad. By one of them, the entire section of the heading is taken out in a single operation. To accomplish this, the cross-head and arms, on the central spindle, are replaced by a very flat, cone-shaped head, which carries a number of individual cutter bits, arranged in diametral lines on the surface of the cone. The circular paths traversed by these bits cover one another, so that the whole mass of coal is broken up.

This modification of the original machine has done good service in some of the bituminous mines of western Pennsylvania, such as those of the Frick Coal and Coke Co. The average speed of advance, under favorable conditions, is 2½ to 3 feet per hour, including the time occupied in moving and setting up after a run. In one case 2,254 linear feet of entry were driven at an average speed of 17 ft. per 9 hours, and an average working cost of 40 cents per foot. In a mine of the Frick Co., 1,475 feet have been driven without a parallel entry for air connections, the exhaust air being discharged backward through an 8-inch pipe. This exhaust assists in ventilating long headings.

For rapid development work, as in opening a bituminous colliery for the "long-wall" method of mining, the Stanley Header is specially advantageous. It has been used, also, by the Col-

orado Fuel Co., with the following results, as compared with hand work:

HAND LABOR.

2 men, 1 10-hr. shift	$4.00
Paid to men for coal produced in driving, 4½ tons @ 50c.	2.25
Cost	$6.25
Distance driven in 10 hours, 3 feet.	

MACHINE WORK.

1 operator, $3.00; 1 helper, $2.50	$5.50
3 shovellers to load coal behind machine @ $2.00	6.00
Compressed air, repairs, depreciation and interest	3.50
Squaring up corners, for timbering and track	5.00
Cost	$20.00
Distance driven in 10 hours, 20 feet.	

Crediting to the machine work the coal produced, viz., 15½ tons @ 50c. loaded, the net cost for the 20 feet of entry was $12.25, or $1.84 per yard. This shows a substantial gain over hand work, both in speed and cost.

Auger Drills.—Though not strictly in place here, reference may be made to the rotary auger drills, for boring holes for blasting in coal, rock-salt and other soft material. They are operated by small compressed air engines—some also by electricity.

Two of these are built by the Ingersoll-Rand Company. The first is of the breast-drill type, similar in general form to a machinist's breast drill. It has a 3-cylinder motor, which can readily be reversed for withdrawing the bit from the hole; total weight, exclusive of the bit, 18 lbs. The second, a heavier machine, is designed to be mounted on a column or bar.

A compressed air auger drill, mounted on single or double column, is made by the Jeffrey Manufacturing Co. Total weight, for a 6 foot vein, 183 lbs. The speed of the engine is about 3,000 revolutions, and that of the threaded feed shaft, 850 revolutions per minute.

Comparison of Coal Cutters.—The chain and disc machines work best in clear coal, of uniform quality and not too hard.

For hard, "bony" coal, or coal containing streaks of pyrites, or "sulphur balls," the pick machines are preferable; because the operator can regulate the strength of the blow and so direct the machine as to cut around a hard place, when necessary. For this reason, however, the pick machine requires more skill on the part of the operator.

Somewhat less slack and fines are made by the disc and chain machines, as the volume of undercut is smaller; but, taking into account the greater flexibility and variety of work possible with the picks, the latter are in many respects advantageous for all-around work. Moreover, in the numerous mines whose product goes chiefly to coke ovens, the larger quantity of fines made by the pick machines is immaterial. Also, in solid, hard coal, the higher undercut of the pick machines causes a more complete breaking up of the whole mass when blasted down, and the coal, therefore, is sometimes more readily loaded.

For chain and disc cutters a fairly good roof is desirable; otherwise props may have to be set so close to the breast or long-wall face as to interfere with the manipulation and shifting of the machines. Coal cutters of all the types can be worked in seams as thin as about 30 inches, or even less; though they are more conveniently operated in seams not less than 36 to 42 inches thick. The continuous-feed chain and disc machines are specially useful for thin, pitching seams in long-wall work, as they will operate with almost equal facility either up or down the pitch.

The "mining rate," or cost of mining by hand, together with the character of the coal seam, will usually determine whether coal cutters can be economically applied in a given mine or district. In general, it may be stated that in seams of average quality and thickness, when the cost of hand mining in the district is not less than 50 to 55 cents per ton (including loading the coal), a saving may be effected by introducing machines.

CHAPTER XXIII

CHANNELING MACHINES

Originally, channeling machines were used almost exclusively for getting out dimension stones in quarry work. Of late years, however, they have been employed in increasing numbers for certain kinds of rock excavation, where it is desired to have smooth, uniform walls; for example, rock cuttings for railroads, canals and water-wheel pits for power plants. They are best adapted for cutting the softer rocks, like limestone, most of the sandstones, slate, shale, etc., though they may be used also for some of the varieties of granite, gneiss, porphyry, schist, and other metamorphic rocks. Hard rocks are best quarried by drilling rows of holes, with wedging or blasting.

In a certain sense channelers resemble reciprocating rock drills, a single bit or a "gang" of bits being attached to the piston rod. But, instead of drilling a series of round holes, the channeler, as its name implies, cuts a continuous, narrow groove, without rotation of the piston and bit. In its typical form, the machine is solidly supported on a heavy carriage or truck, generally mounted on a track laid along the line to be cut. The motive power may be compressed air, electricity, or steam. By means of an auxiliary engine and worm gearing, the whole machine, while at work, is fed forward automatically at a suitable speed. The general construction of a standard compressed air-driven channeler will be understood by reference to Fig. 185. Another design, a track channeler for cutting marble, with adjustable mounting and a reheater mounted on the carriage, is shown in Fig. 186.

General Construction. The construction of channeling machines is varied to suit the conditions of work: 1. For cutting vertical channels only, the rigid head machine is used; that is, the standard supporting the cylinder and accessories is non-adjustable,

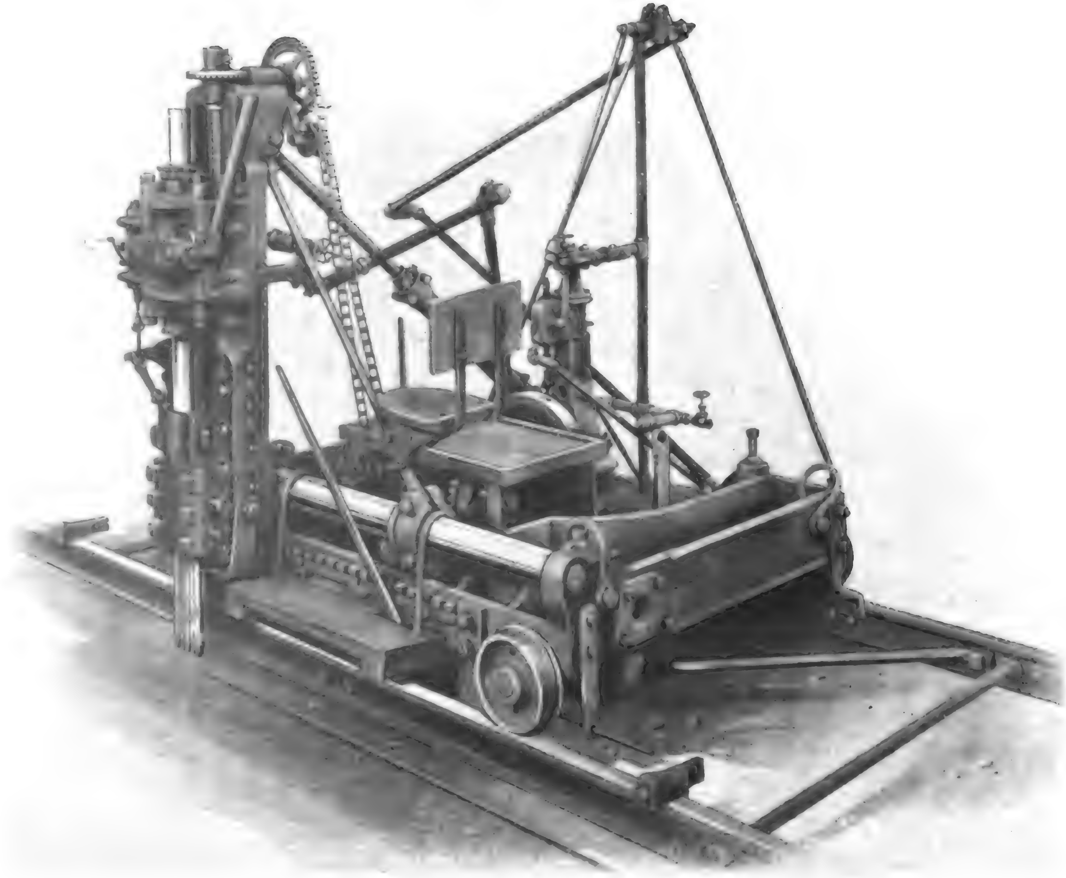


FIG. 185.—Sullivan Track Channeler.

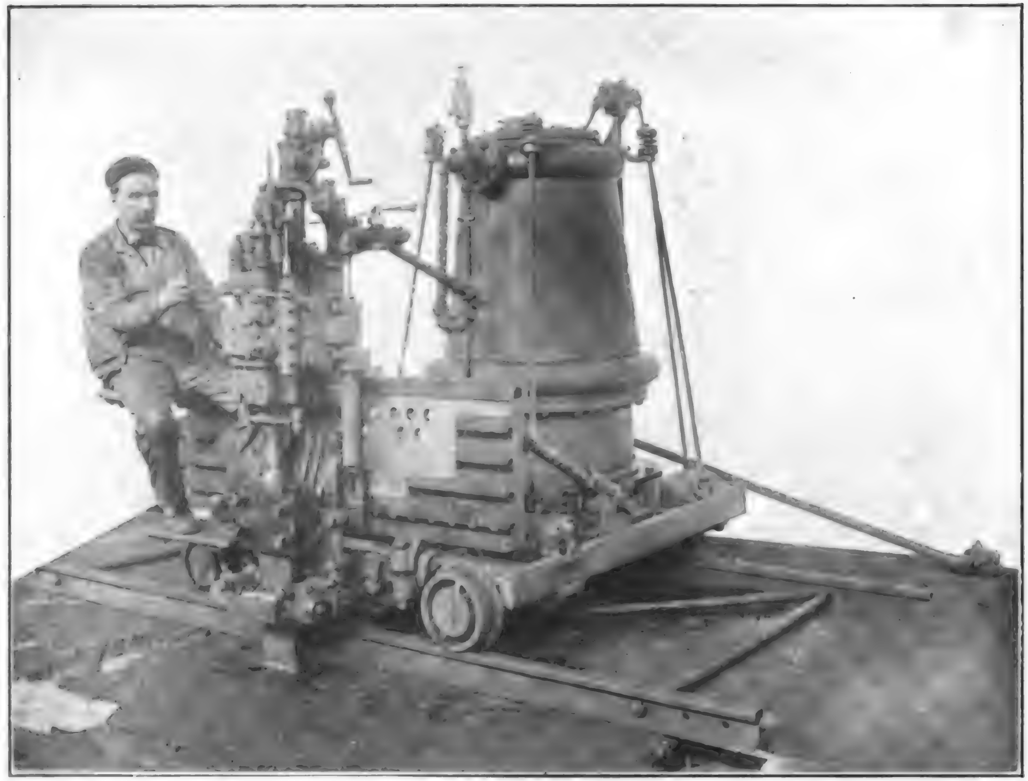


FIG. 186.—Ingersoll-Rand Ram Track Channeler, for Marble.

being permanently fixed in an upright position. Fig. 187 shows a steam-driven channeler of this class. It is employed for canal and railroad cuttings, general rock excavation and for quarrying

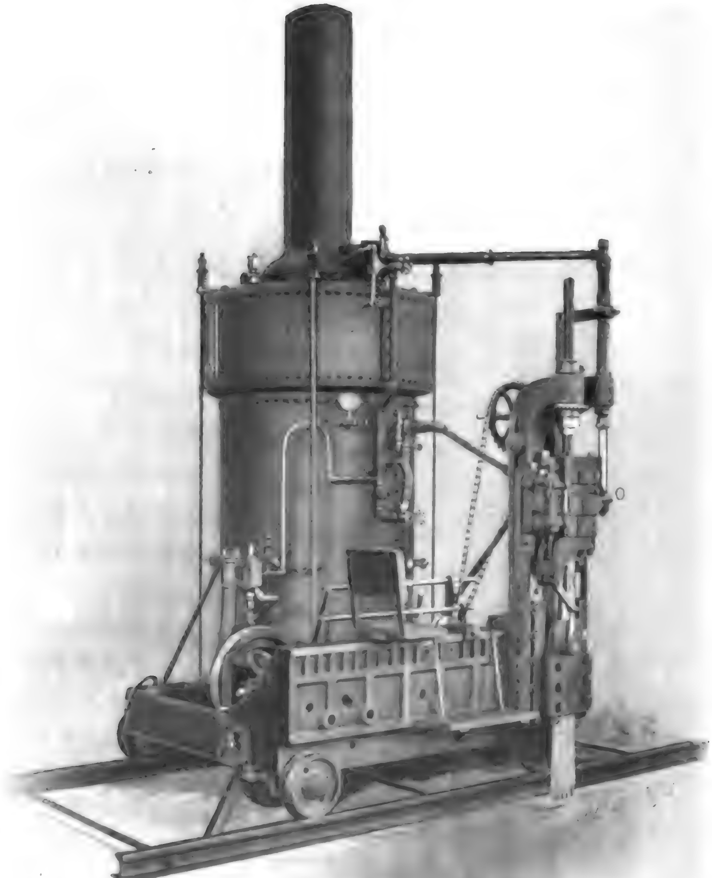


Fig. 187.—Sullivan Rigid Back, Steam-Driven Channeler.

where the strata are horizontal or nearly so. 2. For quarrying building stone lying in inclined stratified beds, like most limestones, the channels must generally be cut at right angles to the

bedding planes; hence, the mounting of the cutting engine is adjustable, for making a channel at any desired angle to the vertical. The cylinder with its appurtenances is swivelled on its supporting frame or standard; or the frame may be provided

FIG. 188.—Sullivan Adjustable Back Air-Driven Channeler.

with T-slots (resembling those of the table of a planer), by means of which the cylinder is bolted firmly in the required position. The supporting frame, in turn (as in Fig. 186, of the Ingersoll-Rand Ram Track Channeler), may be swung back to a nearly horizontal position, for making angle wall cuts along a quarry

face. In different designs, the minimum swing-back angle varies from 15° or 20° to 33° to the horizontal; but it is not often necessary to cut with these machines at less than 45°. Fig. 188 shows a Sullivan channeler of this class. 3. A third form, designed specially for making horizontal channels, or "undercuts," is used less frequently than the others. An Ingersoll-Rand machine of this type is shown in Fig. 189. As indicated in the cut, the head may be bolted to either end of the carriage. 4. Lastly, a light machine, which is in effect a large rock drill, may be mounted on a "quarry bar"—a long, hollow bar, supported at each end by a pair of inclined legs.* This is generally used for drilling a row of holes placed close together, the partitions between which are afterward cut out by a "broaching" bit. Within a few years, a modification of the quarry-bar machine has been brought out by the Ingersoll-Rand Company. It is a true channeler, mounted on a heavy swivel plate, which slides on a pair of horizontal bars, about ten ft. long, supported by inclined legs (Fig. 190). The whole machine is fed along the bars automatically by a small, 3-cylinder engine, which actuates a traveling feed-nut, engaging with a threaded shaft between the bars.

There are many variations in construction in the above-mentioned classes of channeler, to adapt them to local conditions. Among other machines, of entirely different design, may be mentioned the Wardwell and the Bryant channelers. The Wardwell, a heavy machine operated by steam only, has been in successful use for many years. It is intended for making vertical channels, a gang of bits being set in a massive frame, which is given an up and down movement, something after the manner of a jumper drill.

For the first three classes of channeler a gang of from three to five bits is employed. These have long square shanks and are set closely side by side, the cutting edges being alternately at right angles and at 45° to the direction of the channel. This arrange-

* In one of the Sullivan models, the legs are replaced by vertical standards, each carried on a small wheeled truck.

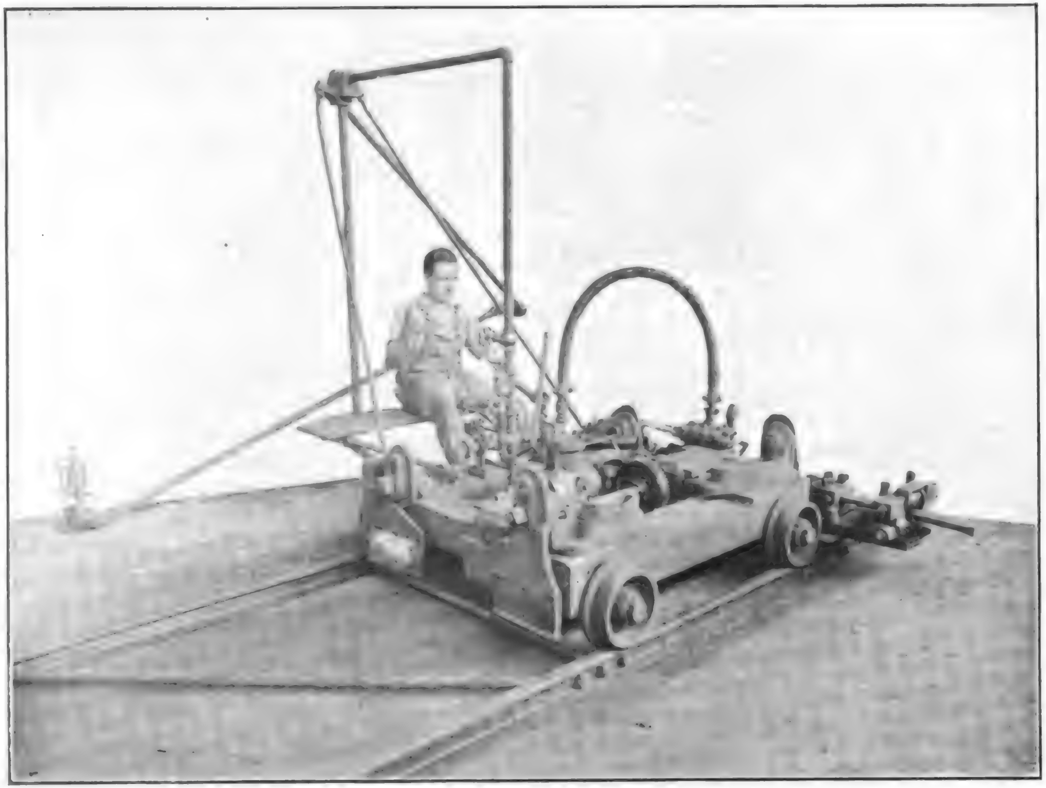


FIG. 189.—Ingersoll-Rand Undercutting Track Channeler, Type H F-3.

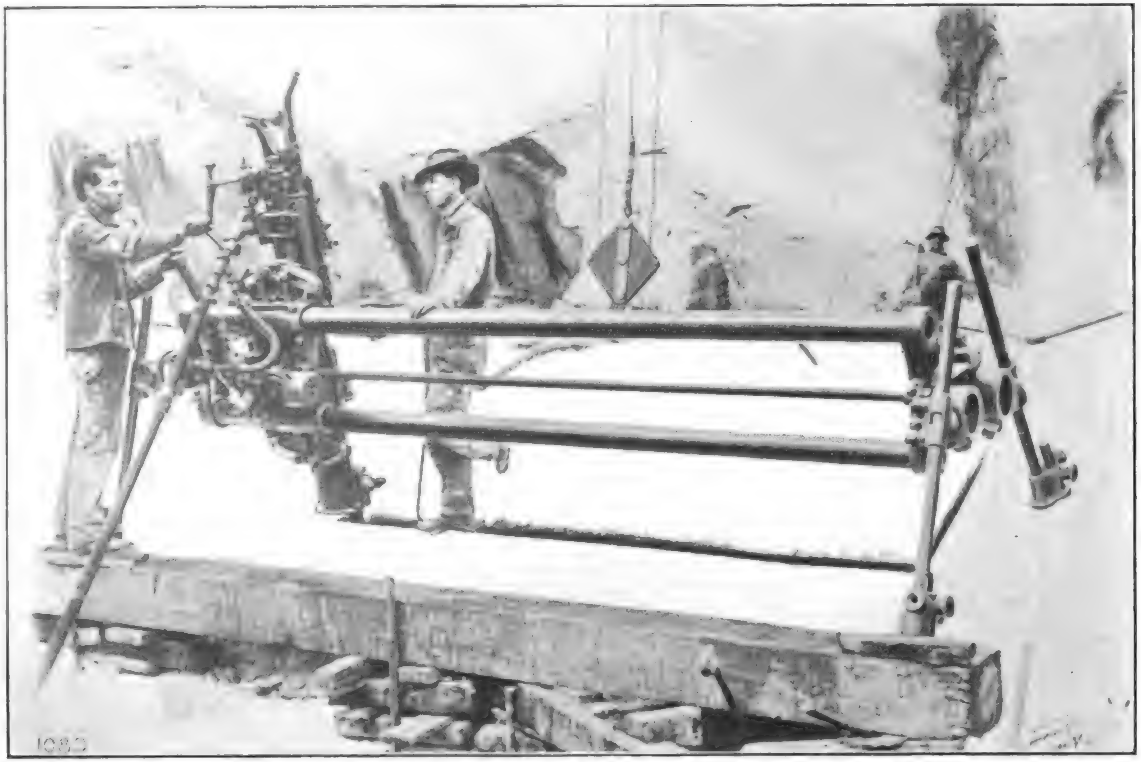


FIG. 190.—Ingersoll-Rand "Broncho" Channeler.

ment forms practically a succession of Z-shaped bits, and insures the cutting of a regular channel with smooth walls. The bits are clamped firmly in a heavy chuck, attached to the piston rod of the engine, and are guided either by a cross-head or (as in some of the Ingersoll-Rand patterns) by a pair of roller guides. Some saving in power has been realized by the introduction of the roller guides; they eliminate part of the weight due to the cross-head, which must be lifted at each stroke, and the friction loss is reduced. The Sullivan Company builds a duplex or double-head machine. There are two cylinders, side by side on a heavy frame, each with its gang of bits and operated by a single valve-chest. As the blows alternate, one piston making its down stroke, while the other is on the up stroke, the machine can be run at high speed without excessive vibration. Its working capacity is correspondingly greater than that of the single-cylinder machines. When the plant consists of a few machines only, they may be advantageously driven by steam (Fig. 187); but, for large-scale work, a higher degree of economy results from the employment of compressed air, furnished by a central plant. Each machine is then provided with its own reheater,* mounted on the carriage (Fig. 186). Air pressures generally range from 85 to 110 lbs.

While at work the main cylinder of the channeler is raised and lowered in its guide shell by a screw-feed, operated automatically or by hand. The hand feed is rarely employed except for the smaller machines. The automatic feed may be caused either by an independent engine, similar to that used for the longitudinal feed of the quarry-bar machine, already referred to; or, by a chain and sprocket drive, from the engine which furnishes the propelling power along the track. The chain feed, as used in the Sullivan channelers, is shown in Figs. 185 and 187. Most of the Ingersoll-Rand channelers are provided with the independent feed engine, which is of the 3-cylinder type, very small and compact in design. In either case, when the cut has reached the required depth, the feed is reversed and the entire

* See Chapter XIX, on Reheaters.

head, with its accompanying parts, is raised preparatory to making the next cut.

Depth of Cut and Speed of Work. The heaviest channelers—those with rigid back or standard—will cut to depths of from 8 to 15 or 16 ft., according to the character of the stone; the swing-back and bar machines will cut from say 6 to 10 or 12 ft., and undercutting machines up to 7 ft. For starting a channel, the width of a bit is from $1\frac{1}{2}$ to a maximum of 4 inches, depending on the depth of cut to be made and on the nature of the stone. The gauges of the successive bits are generally reduced by $\frac{1}{16}$ inch each, the finishing bits usually cutting a width of $1\frac{1}{8}$ inch.

The cutting capacity of channelers varies greatly. It is largest in the softer stones, when of uniform texture and quality, and in fully developed quarries, where the work is systematic and the stone lies below the zone of weathering and surface disintegration. In sandstone of average hardness and under favorable conditions, from 250 to 300 sq. ft. of channel may be cut per 10 hours by the heavy machines; or, including all stoppages and delays, from 4,000 to 4,500 sq. ft. per month; in the softer sandstones and limestones higher duties are obtainable. The swivel-head and other adjustable channelers are lighter than the fixed-back machines and in the same kind of stones their rate of work is generally slower. Machines working in rather hard marbles, like those of Rutland, Vt., will cut from 2,300 to 2,500 sq. ft. per month, or an average of 85 to 100 sq. ft. per day. A single day's work, however, will often greatly exceed these figures. In hard marble or limestone, the smaller bar machines will cut an average of say 40 sq. ft. per 10 hours and up to 125 sq. ft. in softer stones. For hard gneiss, or schist, like that of New York island, an average duty would be 65 to 70 ft. per day.

Tables XLIX and L, showing dimensions, weights, and other data, of the channelers of two well-known builders, will further illustrate the features of these machines.

Recently, the Ingersoll-Rand Company have applied the principle of their "Air-Electric" rock-drill in the design of the cylinder and air compressing mechanism of a track channeler (Fig. 191).

TABLE XLIX

SPECIFICATIONS OF INGERSOLL-SERGEANT CHANNELERS

Size and Type.			Fixed Back Channeler. "H8"	Fixed Back Channeler. "H9"	Swing Back Channeler. "H9"	Swing Back Channeler. 6 in.	Swing Back Channeler. 5 in.	Under-cutting Channeler.	Broncho Channeler.
Diameter of Cylinder		in.	8	7	7	6	5	3½	3½
Length of Stroke		in.	9	9	9	6½	6½	7	6½
Distance of Cut from Vertical Wall		in.	7¼	7¼	7¼	7¼	5	8½ (lift)	
Distance from Center to Center of Cut with Machine Reversed		ft. in.	7 - 0	6 - 0⅝	6 - 8⅝	4 - 7⅜	4 - 6⅝		
Inside Gauge of Track		ft. in.	5 - 3	4 - 4⅝	4 - 4⅝	3 - 0⅝	3 - 0⅝	4 - 0⅜	
Length over all		ft. in.	5 - 3	5 - 3	5 - 2	5 - 5	5 - 5	5 - 10½	14 - 0
Width over all	Without Boiler	ft. in.	7 - 1	7 - 0½	7 - 4½	5 - 5	5 - 2	8 - 3	2 - 6
	With Boiler	ft. in.	7 - 6½	7 - 6	7 - 10				
	With Reheater	ft. in.	7 - 3½	7 - 3	7 - 7				
Height	Without Boiler	ft. in.	7 - 4	7 - 4	7 - 2	6 - 10½	6 - 10½	2 - 10	6 - 0
	With Boiler	ft. in.	10 - 0	10 - 0	10 - 2				
	With Reheater	ft. in.	7 - 4	7 - 4	7 - 2				
Weight of Channeler alone	Without Boiler	lbs.	9,000	9,000	8,000	5,150	4,850	6,800	2,375
	With Boiler	lbs.	12,900	12,900	11,900				
	With Reheater	lbs.	10,300	10,300	9,300				
Total Shipping Weight with Track and Equipment	Without Boiler	lbs.	† 13,900	† 13,900	13,700	† 10,500	† 10,200	† 11,800	3,500
	With Boiler	lbs.	† 17,875	† 17,875	17,675				
	With Reheater	lbs.	† 15,175	† 15,175	15,000				

* Height is from top of rail to top of boiler hood which does not include stack.

† These weights are for domestic shipment. Add 1,000 lbs. for foreign shipment.

TABLE L

SULLIVAN STONE CHANNELERS—SIZES, SPECIFICATIONS AND WEIGHTS

Size.	Type.	Diameter of Cylinder, Inches.	* Height of Machine.		Length Along Track.		Width.		Gauge of Track, Inside Measurement.		Distance, Center of Cut to Wall, Inches.	Distance between Centers of Cuts when Machine is Turned on Track.		Free Air Consumption at 100 lbs. Pressure, Cu. Ft. per Min.†	Weight in Pounds. Machine only.	Weight in Pounds. Equipment Only.	Weight in Pounds. Total.
			ft.	in.	ft.	in.	ft.	in.	ft.	in.		ft.	in.				
Y-8	Rigid head with boiler	8	9	11	6	6	6	10	4	11	7	6	8		13270	6050	19320
Y-8	Rigid head without boiler	8	8	3	6	6	6	10	4	11	7	6	8	400	‡13900	5830	19730
Y-8	Double head without boiler	8	8	3	6	6	6	10	4	11	7	6	8	750	16500	6800	23300
Y	Rigid head with boiler	7	9	11	6	6	6	9	4	11	6	6	6		12235	6050	18285
Y	Rigid head without boiler	7	7	6	6	6	6	9	4	11	6	6	6	300	‡12860	5830	18690
Y	Double head without boiler	7	7	6	6	6	6	9	4	11	6	6	6	550	10230	6800	17030
6½	§ Rigid head with boiler	6½	9	11	6	6	6	10	4	11	9	6	3		10520	5435	15955
6½	§ Rigid head without boiler	6½	6	3	6	6	6	7	4	11	9	6	3	230	6520	5215	11735
Z	Swivel head with boiler	7	9	11	6	6	6	9	4	11	6	6	6		13000	6050	19050
Z	Swivel head without boiler	7	7	6	6	6	8	5	4	11	6	6	6	300	9000	5830	14830
Z	Double head without boiler	7	7	6	6	6	6	5	4	11	6	6	6	550	11210	6800	18010
VW	Swivel head duplex with boiler	6½	10	3	6	6	6	2	4	11	8	6	6		13900	8600	22500
VW	Swivel head duplex without boiler	6½	6	7	6	6	6	7	4	11	8	6	6	500	8900	8575	17475
6½	Swivel head with boiler	6½	9	11	6	6	6	9	4	11	6	6	6		11000	5430	16430
6½	Swivel head without boiler	6½	6	8	6	6	6	5	4	11	6	6	6	230	7200	5215	12415
6½	Double head without boiler	6½	6	8	6	6	6	5	4	11	6	6	6	410	8900	6080	14980
VX	Swivel Head without Boiler. § Standard on bar to cut from vertical to horizontal	4½	5	10	4	3	4	5	3	3	4½			190	2600	3140	5740
VX	Swivel Head without Boiler. § Standard removed from bar and hung at either end of frame for undercutting	4½	2	6	variable				3	3	Height of bench 8½ in.			190	2600	3340	5940

* Machines with boiler are measured to the top of the smoke bonnet without the stack; machines without boiler, to the top of the standard, when vertical. † Without reheating.
‡ Includes three balance weights of 4635 pounds. § These channelers are not equipped with the power hoist.

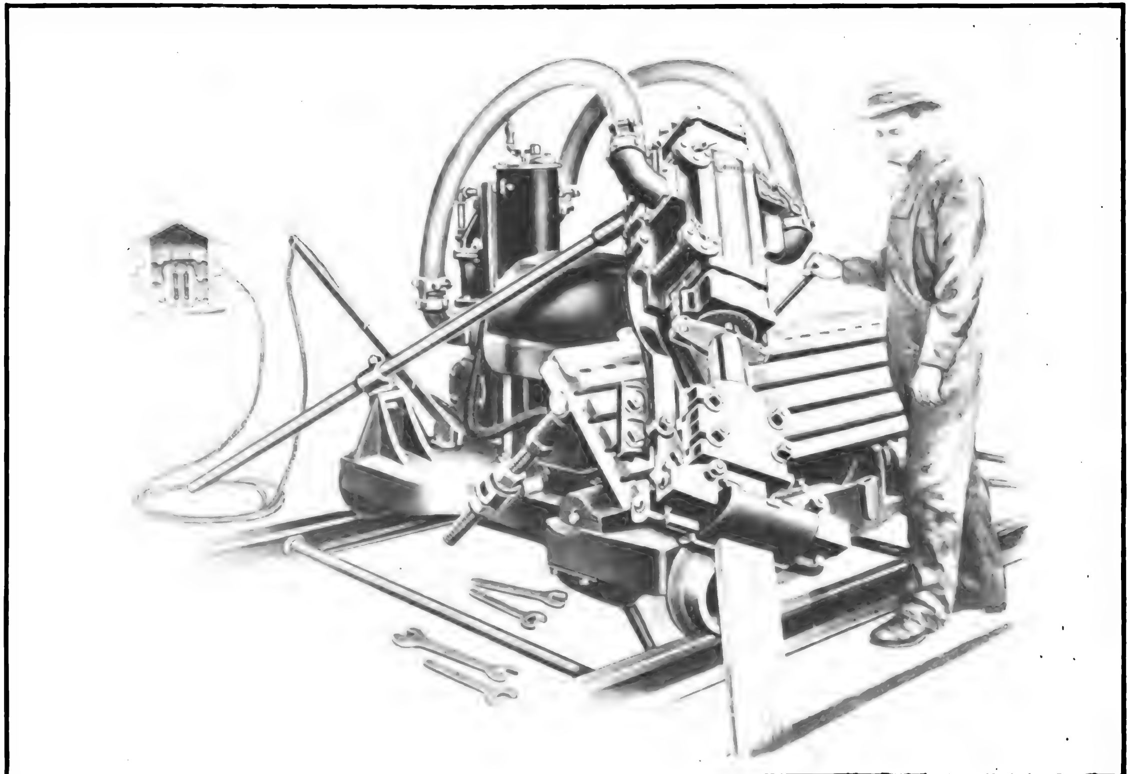


FIG. 191.—Gibson-Ingersoll "Electric-Air" Track Channeler, with Swing Back.

That is, an electric motor, mounted on a carriage, drives a single-acting air compressing pulsator, which is connected to the channeler cylinder in a manner similar to that of the Temple-Ingersoll rock-drill, described in Chapter XX. For each double stroke of the pulsator, there is a blow and return stroke of the channeler piston; the speed of stroke being thus controlled by the speed of the electric motor which furnishes the power. Favorable results, both as to power cost and maintenance, have already been secured. This channeler has a swivel head and a swing-back support, and is therefore suitable for varied quarry service.

CHAPTER XXIV

OPERATION OF MINE PUMPS BY COMPRESSED AIR

It is intended here to deal only with that part of the extensive subject of mine drainage which has to do with the employment of compressed air as a motive power. Under this head there are three general forms of apparatus:

1. Direct-acting pumps, single-cylinder, duplex, or compound.
2. The air-lift pump.
3. Pneumatic displacement pumps.

In this chapter the first class only will be considered.

Simple, Direct-acting Pumps. Notwithstanding the general similarity in the behavior of steam and compressed air, when used in the cylinders of direct-acting pumps there are some important points of difference. By first considering briefly the construction of the types of pump in common use the results obtainable from the employment of compressed air can best be set forth.

The development of the direct-acting pump dates from Henry R. Worthington's invention in 1841; and the greater part of all the pumping in the mines of this country, and much of it in other countries also, is done by pumps of this class. The cylinders are set tandem, the power being transmitted from the steam to the water cylinder through a piston-rod common to both. As there are no rotating parts, the length of stroke is controlled by the admission and exhaust of the steam. In all the simple pumps the valve motion involves the use of an auxiliary valve, whose movements are governed by the reciprocating movement of the piston, and which in turn operates the main valve. The duplex form consists essentially of two simple pumps, set side by side, with an inter-dependent valve motion; that is, the valve of each is operated positively, through a system of levers, by the movement of the piston of the other side.

Though direct-acting pumps are strong and reliable, simple in construction, and occupy but little space, they are extremely uneconomical machines, unless the steam cylinders are compounded. It is hardly necessary to say that this ought not to be the case. Pumping is an operation that should be conducted economically, especially in connection with mining, where the pumping of water is classed as "dead work." Moreover, the conditions in themselves are not unfavorable. A pump works under a practically constant load, from the beginning to the end of each stroke, the only necessary variation—which need not be large—occurring at the instant the discharge valves open.

The trouble is that, in attaining compactness, simplicity, and moderate first cost, the power is not applied in simple, direct-acting pumps to the best advantage. As there is a constant load, but no fly-wheel to equalize the power, steam must be admitted at full pressure throughout the entire stroke; otherwise the piston would be unable to reverse, and would come to a standstill. Such a pump must work practically without cut-off, and therefore a cylinderful of steam, nearly at initial pressure, is exhausted at each stroke. In some pumps the terminal pressure is quite as high as the initial. A duplex, non-compound pump, having a positive valve motion, may at times be even a more extravagant steam-consumer than a single-cylinder pump, since one piston may reach the end of its stroke before the other is ready to reverse its valve. In such case the momentum of the incoming steam fills the cylinder at initial pressure at the moment of exhaust.

For steam-driven pumps there are several ways of improving these conditions:

1. The adoption of compound or triple expansion cylinders. This type is suitable for the larger sizes of pump, and its use is increasing for mines whose depth and quantity of water warrant the higher first cost. The space occupied is but little greater than for simple pumps of the same capacity, and satisfactory results are obtained when they work under proper conditions and with sufficient initial pressure.

2. While retaining the tandem form, a fly-wheel may be introduced, driven from the cross-head or from the steam-cylinder connecting-rod. This is a reversion to a type of pump long ago discarded for general service in this country, in favor of the simpler but less efficient form with no rotating parts. Although such a pump occupies much more space and its first cost is increased, there can be no doubt as to the advantages of being able to use the steam expansively, without the necessity of compounding. A large number of pumps of this description are now employed in mines; many of the Riedler pattern and some of less elaborate and expensive design, such as the Prescott and others, in which an early cut-off—at one-quarter or even one-eighth stroke—is satisfactorily adopted.

Notwithstanding the advances made along these lines in the mechanical engineering of pumps and the added economy gained in their operation, it has been very generally assumed in the past that similar economies are not attainable when compressed air instead of steam is employed as the motive power. Yet the advantages accruing from the utilization of compressed-air transmission in mines are marked. As the heavy losses due to radiation and the condensation of steam in pipe-lines are avoided, the transmission of power by compressed air may be conducted with a high degree of efficiency. No difficulty exists as to the disposal of exhaust steam underground, nor is any danger to be apprehended from the rupture of a compressed-air pipe, while the bursting of a steam pipe in a shaft or in the mine workings may cause serious trouble. The failure to realize these advantages, and the unsatisfactory results obtained in most cases from compressed-air-driven pumps, are due largely to the fundamental differences in the behavior of steam and compressed air when used in a motor cylinder. In Chapter XVII reference has been made to the reduction of cylinder temperature accompanying the expansion of compressed air. The point of cut-off being the same, this causes lower terminal and mean pressures with air than with steam. In other words, at a given initial pressure and without reheating, a cylinderful of air develops less power.

This property of air, together with the fact that it does not condense, indicates clearly that steam and compressed air are not equally well adapted for use in an engine of the same design. It is not easy to understand, therefore, why mechanical engineers and especially pump-builders have not given more attention to the production of pumps properly designed for the use of compressed air. Few, if any, other branches of motor-engine practice have been so neglected. Lack of information among users of compressed air is responsible in part; in addition to which it is not generally realized that relatively unimportant modifications, at small cost, would produce much better results. Users of the ordinary steam pump have become accustomed to its low economy, and, because it is strong and serviceable, it is apt to be accepted without question when compressed air is used instead of steam. But in applying compressed air to the inefficient single-cylinder pump, as usually designed for steam, the net result is no better, and may be even worse, than that obtained from steam. The clearance spaces are large and, as the air is admitted to the cylinder throughout full stroke, it is used in a wasteful manner. Moreover, the stroke is often shortened by imperfections in the valve action.

Another unfavorable feature of mine pumps driven by compressed air is the frequently improper selection of the cylinder proportions and arrangement of the plant. In mines having a number of levels the pumps are distributed according to varying requirements as to height of lift and quantity of water to be raised. The lowermost pump may have to work under a heavy head; others under a head of only 100 or 200 feet. As all are usually operated from the same pipe line and under a common air pressure, it is clear that the dissimilarity of working conditions must be met by proportioning the water and power ends of each pump according to the work to be done. But, through error or carelessness, the power end is often badly out of proportion, the tendency being to err on the side of furnishing too much power. The steam (or air) cylinder may be of such size as to require a pressure of only 30 or 40 lbs. per sq. in., while the pipe-line press-

ure is 70 or 80 lbs., as usual with mine compressor plants. So it often happens that the deepest pump in the mine is the only one operating under a proper pressure. The cylinders of the others, even if running under throttle, are filled with air at full pressure when exhaust takes place.*

The difficulty with common direct-acting pumps is thus two-fold: the air is used without expansion, and the pressure is often higher than is necessary. Recognizing, however, the convenience with which the inexpensive, ready-made single-cylinder pumps may be installed, and that in many cases efficiency of operation is really a secondary consideration, a few points will here be discussed as to their employment, and the volume of air required for a given quantity of work. Questions relating to the expansive use of compressed air for pumps will be taken up afterward.

Cylinder Dimensions of Simple Pumps. In calculating the sizes of cylinders for a simple, or single-cylinder pump, to work under given conditions, the dimensions of the water cylinder must first be determined. There are three variables to be dealt with, *viz:* diameter, length of stroke, and number of strokes per minute; or the last two factors named may be combined in the shape of piston speed per minute. The volume of water to be raised being given, the cylinder dimensions may be obtained from lists of standard sizes of pumps, which would usually be adhered to on the ground of saving in first cost. With a given air pressure and head of water, the diameter of the air cylinder obviously depends upon that of the water cylinder. The following relation between the two has been determined by Mr. William Cox:† "Area of air cylinder is to area of water cylinder as half the head is to the air pressure." By the same writer a ready reference table has been constructed, covering the air pressures generally used for common, direct-acting pumps:

* Some suggestive remarks on this subject are made by Frank Richards, "Compressed Air," pp. 171–172.

† *Compressed Air Magazine*, Feb., 1899, p. 583. (By permission.)

TABLE LI

RATIOS OF DIAMETER OF AIR CYLINDER TO DIAMETER OF WATER CYLINDER

Head in Feet.	Air Pressure, Pounds.						
	20	25	30	35	40	45	50
50	1.12	1.00	0.91	0.84	0.79	0.74	0.71
100	1.58	1.41	1.29	1.20	1.12	1.05	1.00
125	1.77	1.58	1.45	1.34	1.25	1.18	1.12
150	1.94	1.73	1.58	1.45	1.37	1.29	1.22
175	2.09	1.87	1.70	1.58	1.48	1.39	1.32
200	2.24	2.00	1.82	1.69	1.58	1.49	1.41
225	2.37	2.12	1.94	1.79	1.68	1.58	1.50
250	2.50	2.24	2.05	1.90	1.77	1.67	1.58
275	2.62	2.35	2.14	1.98	1.85	1.75	1.66
300	2.74	2.45	2.24	2.07	1.94	1.82	1.73
325	2.85	2.55	2.33	2.16	2.02	1.90	1.80
350	2.96	2.64	2.42	2.24	2.09	1.97	1.87
375	3.06	2.74	2.50	2.31	2.16	2.04	1.94
400	3.16	2.83	2.58	2.39	2.23	2.11	2.00
425	3.26	2.92	2.66	2.46	2.30	2.17	2.06
450	3.35	3.00	2.74	2.53	2.37	2.24	2.12
475	3.44	3.08	2.82	2.60	2.44	2.30	2.18
500	3.53	3.16	2.89	2.67	2.50	2.36	2.24

Ratios for intermediate heads and pressures may be obtained by interpolation.

In this table the unit diameter of water cylinder is taken as one inch. Diameters of air cylinders, as calculated, will be in decimals, and often of odd sizes not occurring in practice. After determining the exact diameter, the nearest standard diameter of cylinder would be chosen and the air pressure and piston speed adjusted accordingly.

Volume of Air for Pumps Working without Expansion. To determine the volume of free air required to operate a direct-acting, single-cylinder pump, working without cut-off, the formula here given will be found convenient:*

$$V = 0.093\ W_2 \frac{h \times G}{P}, \text{ in which:}$$

V = volume of free air in cubic feet per minute.

h = head in feet under which the pump is to work.

* Ibid., p. 581.

G = gallons of water to be raised per minute.
P = receiver gauge pressure of air to be used.
W_2 = volume of free air corresponding to one cubic foot at the given pressure, P.

In this formula, which is based on a piston speed of 100 feet per minute, fifteen per cent. has been added to the volume of air to cover losses. The following table, giving values of W_2 and $0.093\ W_2$ for different pressures, may be used in connection with the formula:

TABLE LII

Air Pressure P, in Pounds.	W_2	$0.093\ W_2$
15	2.02	0.18786
20	2.36	0.21948
25	2.70	0.25110
30	3.04	0.28272
35	3.38	0.31434
40	3.72	0.34596
45	4.06	0.37758
50	4.40	0.40920
55	4.74	0.44082
60	5.08	0.47244
65	5.42	0.50406
70	5.76	0.53568
75	6.10	0.56730
80	6.44	0.59890
85	6.78	0.63054
90	7.12	0.66216

For example, let it be required to find the volume of free air per minute required to raise 200 gals. of water to a height of 150 ft., the gauge pressure being 30 lbs. From the table, $0.093\ W_2$, corresponding to 30 lbs. $= 0.2827$; hence,

$$V = 0.2827 \times \frac{200 \times 150}{30} = 282.7 \text{ cu. ft. free air.}$$

The horse-power may be calculated from Table LIII, in which the mean pressures per stroke (from Table VII), for the different terminal pressures, are given in the second column, and the calculated horse-powers per cubic foot of free air used, in the third column:

TABLE LIII

Terminal Pressure, Pounds.	Mean Pressure per Stroke.	Horse-Power per Cubic Foot Free Air.
20	14.40	0.0628
25	17.01	0.0743
30	19.40	0.0847
35	21.60	0.0943
40	23.66	0.1033
45	25.59	0.1117
50	27.39	0.1196
55	29.11	0.1270
60	30.75	0.1340
65	32.32	0.1406
70	33.83	0.1468
75	35.27	0.1527
80	36.64	0.1583

As the horse-power corresponding to a given terminal pressure does not increase in constant ratio with the initial air pressure, it follows that the higher pressures are not so economical for simple pumps as low pressures. Expressed in another way, the work of compression decreases with the air pressure, and therefore the useful work done in a pump using air at full pressure is greater at low pressures and its efficiency is increased. Thus, in the example given above, the horse-power developed in using the 282.7 cu. ft. of free air, at a pressure of 30 lbs., is:

$$282.7 \times 0.0847 = 23.94 \text{ h.-p.}$$

If the air pressure employed were 50 lbs., the cu. ft. of free air would be 245.52 and the corresponding h.-p., 29.36, the added power cost being 5.42 h.-p. It may be stated that the difference in favor of the lower air pressure is offset in part by the fact that, at the higher pressure, a pump with a smaller power cylinder will do the same work, thus saving in the first cost.

But the low pressures thus shown to be suitable for simple pumps would not serve for machine drills, which must be considered first, as they are in nearly all cases the chief users of compressed air in mines and quarries. To secure the best results from the pumps, a separate, low-pressure compressor would be required, a provision which is usually out of the question. Since

it is generally necessary to use high-pressure air, at, say, eighty or ninety pounds gauge, the air must either be wire-drawn into the pump cylinder or else reduced to the required pressure before being delivered to the pump.

In the first case, the results as to volumes of air used, as given in the preceding discussion and tables, must be modified by introducing a factor of increase, based on the ratio which the pressure to be used in the pump bears to the pressure carried in the air main. Edward A. Rix furnishes a table,* part of which is abstracted in Table LIV. It shows the volumes of free air theoretically required for a unit of 10,000 ft.-gals. of work (=83,000 ft.-lbs. or 2.5 h.-p.), at different air pressures, together with the actual air consumption and horse-powers, all referred to a standard receiver pressure of 90 lbs.

TABLE LIV

Gauge Pressure, Pounds.	Ratio of Compression, Referred to 90 Pounds.	Cubic Feet of Air Calculated from Cox's Formula.	Factor of Increase for Wire-Drawing from 90 Pounds.	Increased Volume, Cubic Feet.	Actual Horse-Power at 90 Pounds.	Efficiency on Basis of 2.5 Horse-Power Theoretical.
20	3.	113	1.26	142	28.6	9
25	2.6	108	1.22	125	25.	10
30	2.3	97	1.19	115	23.	11
35	2.1	93	1.17	108	21.5	11.6
40	1.9	89	1.14	102	20.5	12.2
45	1.7	87	1.12	97	19.7	12.7
50	1.6	85	1.11	93	19.	13.1
55	1.5	82	1.09	89	18.2	13.7
60	1.4	80	1.07	86	17.4	14.3
65	1.31	79	1.06	84	16.8	14.9
70	1.24	78	1.05	82	16.4	15.3
75	1.17	77	1.04	80	16.	15.6
80	1.1	76	1.03	78	15.6	16.
85	1.05	75	1.02	76	15.2	16.4
90	1.0	74	1.0	74	14.8	16.9

The factors in column 4 are assumed as about 70 per cent. of the ratios of the absolute temperatures due to expansion of the air from 90 lbs., to the air pressures in column 1. They may be taken to apply when the length of air main from the compressor

* *Transactions Technical Society of the Pacific Coast*, Aug. 3d, 1900.

to the pump is moderate, as in carrying the air to a pump situated at the bottom of an ordinary shaft. The showing is a poor one, but the unfavorable working conditions, as to the type of pump and mode of using the air, must be taken into account.

In the second case, the normal air pressure carried in the mine (say, ninety pounds) may be reduced to a suitable pump pressure by placing a reducing valve in the air main. The increase of volume thus produced will be accompanied by a considerable drop in temperature, so that the full increase is not realized. Part of the lost heat will be regained by friction, and from external sources if there be any considerable length of pipe between the reducing valve and pump; but the efficiency will be materially increased if the cold, partly expanded air be passed first into an underground receiver and thence to the pump. This arrangement has been satisfactorily adopted, for example, in the case referred to at bottom of p. 277. An adjustable spring-reducing valve is set to furnish any desired pressure below that in the main. That is, the volume of air allowed to pass is such as to maintain automatically a certain difference in pressure between that in the main and the pipe leading to the second receiver. The latter serves three purposes: (1) if it be of ample size or of the tubular type the air will regain nearly, if not quite, its normal temperature; (2) much of the entrained moisture will be deposited, and trouble from freezing avoided; and (3) the receiver, if placed near the pump, will minimize the pulsations and equalize the air pressure.

In the particular instance to which reference is here made, two underground receivers were installed 300 feet apart, the reducing valve being put in the main just above the first receiver. This arrangement not only caused a very complete deposition of the moisture, but the air entirely recovered its normal temperature by the time it left the second receiver on its way to the pump. The main air pressure was 85 lbs., and at the pump about 45 lbs. Indicator diagrams showed 128.5 horse-power developed by the compressor and 16.45 horse-power at the pump, or an efficiency of 12.5 per cent.; thus agreeing quite closely with

the figures in Table LIV. Subsequently, by compounding one of the pumps, using 62 lbs. initial pressure in the high-pressure cylinder and admitting some live air to the intermediate pipe between the cylinders, the efficiency was raised to 25.9 per cent. This must be considered a fairly satisfactory performance for a pump not specially designed for its work.

By adopting stage compression or by reheating, or both, the total efficiency can of course be increased considerably beyond the efficiencies shown in the table. Mr. Rix states, in his article previously mentioned, that by actual test of a number of simple pumps he has found their work to be approximately 135 ft.-galls. per cu. ft. of free air. For stage compression the efficiency is increased by 15 per cent. (giving, say, 155 ft.-gals.), and by reheating the 135 ft.-gals. is increased by the ratio of the absolute temperatures under which the pump works, without deducting the small cost of reheating.

Prevention of Freezing of Moisture. Though this subject has already been discussed at some length, several additional points may be noted in connection with pumping. Some benefit may be derived by leading a jet of water from the pump column into the air pipe, just before reaching the pump. A very small quantity of water will suffice to prevent an excessive drop in the temperature of the exhaust. A better way is to tap a one-quarter-inch pipe into the column pipe, draw down the end of this pipe to, say, one thirty-second of an inch and insert the nozzle so formed into the exhaust port. The author has observed the plan of carrying a small steam jet close to the exhaust port; but it is obvious that this is feasible only when steam is used near-by for some other purpose. Moreover, steam so applied is utilized much less perfectly than when used in a cylinder jacket. If steam be available, a little may be injected into the feed air pipe near the pump. An intimate mixture between the steam and air is thus produced, and in condensing the latent heat of the steam is given up. If water at 212° F. be injected, each pound in cooling down to 32° F. will give up 180 thermal units. But with steam at the same initial temperature, each pound in condensing gives up 966

thermal units, in addition to the 180 units imparted in cooling to 32°. Still another mode of preventing freezing is to warm the compressed air by passing it through a coil of pipe, placed in an enlarged section of the water column, or else in the pump-suction pipe.

Compressed-Air-driven Compound Pumps. It is a commonly held idea that if compressed air be used for operating compound, direct-acting pumps, it should be employed like steam, with a cut-off in each cylinder. The resulting drop in cylinder temperature would be obviously less than that caused in a single cylinder by the same ratio of expansion from a given initial pressure. But in aiming thus to attain a higher efficiency, by adopting the largest possible range of expansion, very low cylinder temperatures would still be produced. The loss of heat takes place chiefly within the cylinder, instead of in, and just outside of, the exhaust port, as is the case with pumps working at full pressure. Furthermore, though the same total fall of temperature occurs in either case, when the air expands within the cylinder the force of the exhaust is diminished by the low terminal pressure, and the inner portions of the ports are the more liable to be choked with ice.

In order to use the air expansively the necessity for reheating in some form is clearly indicated, aside from any question of gain in economy. Various plans have been tried of warming the cylinders by the application of external heat, such as enveloping them in a hot-air jacket, surrounding them by water, even heating them by the flames of large lamps or torches. But, aside from other objections to such devices, air is too poor a conductor of heat to render these means at all efficient.

The mode of applying extraneous heat may be varied in several ways, *viz:* (1) Preheating the compressed air sufficiently to permit of a reasonably early cut-off in each cylinder, while still avoiding too low an initial temperature in the low-pressure cylinder; (2) in addition to preheating, the air may be reheated between the cylinders; (3) using cold air at full pressure in the high-pressure cylinder and expanding into the low-pressure cylinder,

with or without reheating; (4) using cold air at full pressure in both cylinders, the air being expanded between them, with the application of reheating.

The first two methods are feasible when the compound pump is of suitable design and the heating properly applied; but there would be an undesirable variation in power and speed, for an engine necessarily working under a constant load, if the pump be of the usual direct-acting type, without fly-wheel. Moreover, under the first plan a high initial temperature would be necessary. If the expansion be adiabatic, from an initial pressure of, say, eighty pounds to atmospheric pressure and normal temperature, the temperature to which the air would have to be preheated is given by the expression:

$$T' = T\left(\frac{P'}{P}\right)^{\frac{n-1}{n}} \text{ or, } T' = 70^\circ + 459^\circ\left(\frac{80+15}{15}\right)^{0.29} = 446^\circ\text{F.}$$

Although this temperature would be rapidly lowered during the stroke, proper lubrication of the cylinder might be interfered with. The third method would avoid in part the difficulty of variation in power and speed, though there would still be a variable back-pressure on the high-pressure piston; but the increase in volume due to clearance, and on expanding into the passages and intermediate pipe to the low-pressure cylinder, would considerably reduce the temperature of the air, and a large further drop would ensue during the work of expansion in the low-pressure cylinder. Such temperature drop may be prevented, or at least diminished, by introducing a receiver-reheater between the cylinders, with material gain in efficiency. This method has frequently been adopted, and on the whole is much preferable to the two first mentioned.

The fourth arrangement, however, appears to be the most satisfactory. As has been pointed out by E. A. Rix,* in the practical application of compressed air to pumps only a small part of the total possible work of expansion within the two cylinders can be realized, even in favorable circumstances. Never-

* *Transactions Association of Engineering Societies*, 1900. Mr. Rix also proposes the use of three-, and even four-cylinder pumps.

theless, if properly installed and operated, it becomes perfectly practicable to drive a compound pump by compressed air. It is a much more satisfactory machine than a single-cylinder pump, and is capable of working with a fair degree of efficiency. This may be accomplished by expanding the air between the cylinders only, restoring the consequent loss of pressure by reheating and employing full pressure in both cylinders. Thus no drop of temperature takes place in the cylinders themselves, and the pressures, back-pressures, and speed are constant. Each air card is practically rectangular in shape. The pressure drop between the cylinders may be made small; in fact, it need not be more than is sufficient to give the head necessary to cause an active flow of air into the intermediate reheater and thence to the low-pressure cylinder. A drop of, say, 20 lbs. for an initial pressure of 70 to 80 lbs. will usually answer.

The degree of heat to be imparted by the intermediate reheater, to restore the heat lost by a drop of 20 lbs., would be only 204° F., for a final temperature of 60° at exhaust. If the pump be suitably situated, an ordinary fuel-burning reheater may be employed; or, should this be inadmissible, the water from the pump-suction or column pipe may be utilized for reheating, as already suggested. An example of this arrangement, which has often been cited, is to be found in the Gwin Mine, Calaveras Co., California.* A Worthington compound pump, having a capacity of 200 gals. per minute, was installed on the 600-ft. level of the mine. Placed in the suction pipe of the pump is a 300-horsepower Wainwright heater, with corrugated copper tubes. The water in the pump, at a temperature of 60° to 70° F., passes through the heater tubes on its way to the pump-suction valves. The air, on being exhausted from the high-pressure cylinder, at a pressure of 35 lbs., passes into the heater and through the spaces between the tubes. In this way, the temperature of the air is raised practically to that of the water and, after expanding again in the low-pressure cylinder, is exhausted without freezing. Should the sump water be foul, the heater tubes must be cleaned

* Installed by E. A. Rix. See *Engineering and Mining Journal*, 1905.

from time to time; otherwise the coating of sediment materially reduces their conductivity. Still better results would be obtained from such an installation by employing a fly-wheel pump with a shorter cut-off. The lower temperature could then be met by water-jacketing both cylinders, the jackets being supplied with water by a small pipe from the pump column. Though the quantity of heat thus restored to the expanded air is far smaller than that which would be derived from a fuel-burning reheater, this simple device is convenient and satisfactory for underground service.

By employing reheating in connection with properly designed and operated air-driven compound pumps, efficiencies of 40 to 50 per cent. may be realized. With 3-cylinder pumps, furnished with intermediate heaters, the efficiencies are still higher, reaching even 70 per cent. Reference has already been made to the economic advantages of using the Cummings system of high-pressure transmission for operating compressed-air pumps.

CHAPTER XXV

PUMPING BY THE DIRECT ACTION OF COMPRESSED AIR

THE different modes of raising liquids by the direct pressure of air, without the intervention of a piston or other moving part, embody no new idea, but it is only in quite recent years that they have taken such shape as to render them useful for pumping on a large scale. Besides the fundamental considerations of cost and efficiency of plant, which affect alike all systems of pumping, another question becomes of prime importance in connection with these methods of applying compressed air, *viz:* the practicable limits of depth or head at which they will work. These limits depend on the gauge pressure and mode of using the air. In point of efficiency, several forms of plant included under this head are distinctly inferior to well-designed steam-driven piston and plunger pumps. But when operated under proper conditions and with expansive use of the compressed air, recent modifications and improvements have brought several of them to a very satisfactory degree of efficiency. In first cost they compare favorably with pumps of the usual types, and, because of their large capacity and low maintenance cost, all possess marked advantages for some kinds of service.

There are two classes of pumps in which the principle in question is employed:

1. Pneumatic-displacement pumps, using compressed air with or without expansion.

2. "Air-lift" pumps, working expansively.

Pneumatic-Displacement Pumps. These are of several kinds. In the type form the compressed air is caused to act directly upon the surface of the water contained in a submerged closed

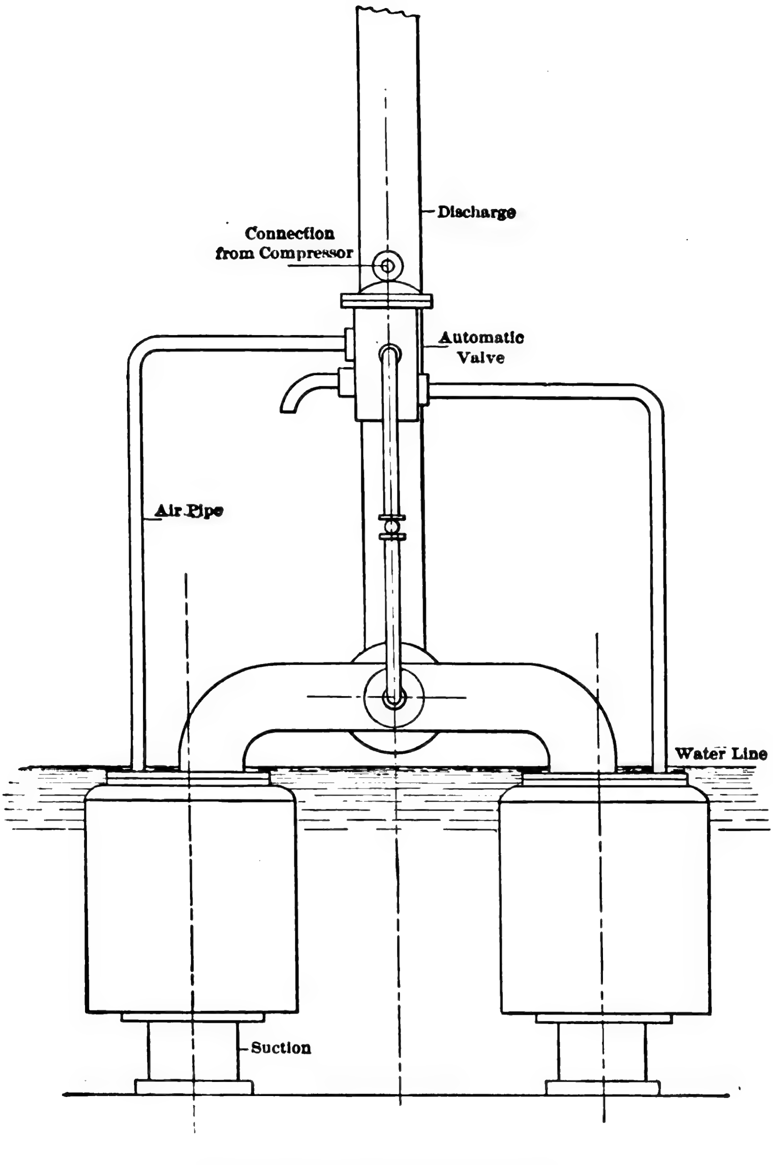


FIG. 192.—Merrill Pneumatic Pump.

chamber or tank, suitable valves being provided for controlling the admission of air and water. As the name implies, the water is displaced by the air and is discharged from the tank through a column pipe. There may be either one or two tanks, the column pipe in the latter case being common to both. With one tank, the flow of water from the pipe is intermittent; with two, practically constant, the pair of tanks then resembling in their relation to each other the chambers of the ordinary steam pulsometer pump. Aside from the simplicity of construction and absence of moving parts subjected to wear, which adapt it for mining, as well as for general service, such as pumping from wells and other sources of water supply, the pneumatic-displacement pump has a distinct advantage for pumping chemical solutions, acids, etc., which would corrode the mechanism of a piston pump. It is evident, however, that the head or pressure under which the ordinary displacement pumps will work is limited absolutely by the air pressure employed.

The double-chamber pump, as built by the Merrill Pneumatic Pump Co., will serve to illustrate details of construction and operation. Fig. 192 is a diagram of this pump, showing the submerged chambers, with their connections to the discharge pipe. Air from the compressor enters a chest through an automatic valve, which opens connection alternately with the two water chambers. The air pressure to be employed depends on the height of lift. Since the weight of a column of water is 0.434 lb. per foot of head, the height to which a given air pressure will raise water is equal to the gauge pressure divided by 0.434; thus, air at 80 lbs. will pump to a height of $\frac{80}{.434} = 184$ ft. In practice, however, to cover friction, leakage, absorption of air by the water, and to provide the necessary dynamic head for overcoming inertia and securing a proper speed of discharge, an additional air pressure is required. In terms of volume, 1 cu. ft. of water will be displaced per cu. ft. of compressed air. One cu. ft. of air at 80 lbs. $= \frac{1 \times (80 + 15)}{15} = 6.33$ cu. ft. free air. To this should be added

for losses, etc., say 20 per cent., making a total of 7.6 cu. ft. free air per cu. ft. of water. Taking 1 gal. of water equal to 0.134 cu. ft., the work done per cu. ft. of compressed air, acting against a head of 184 ft., will be: $\frac{184}{0.134 \times 7.6} = 180$ ft.-gals. $= 1503$ ft.-lbs. In some cases a larger allowance than 20 per cent. should be made. The actual work done in compressing 1 cu. ft. of air to 80 lbs. gauge, by a single-stage compressor (see Table V) is 0.183 horse-power, or 6039 ft.-lbs.; hence, the efficiency of the pump, on the basis of allowance for losses assumed above, is nearly 25 per cent., which compares favorably with the efficiencies of single-cylinder direct-acting pumps.

The displacement pump in its usual form works like a simple piston pump, in exhausting at each stroke a tankful of air practically at gauge pressure. By employing a series of these pumps in a shaft, however, and using the air expansively, it is evident that, with a given initial pressure, the possible height of lift and the total efficiency of the system will greatly exceed that shown above.* This can be done by a suitable valve control, by which the air is expanded from the lowermost tank to the one next above, and so on, for smaller and smaller lifts toward the top of the series. When the last tank is discharged, the whole system is occupied by expanded air, at a pressure of two or three pounds, which is then exhausted into the atmosphere. Air is admitted by the valve at intervals into the lowest tank, and the working of the system proceeds automatically. At 80 lbs. air pressure, water can thus be raised to a height of about 330 ft., instead of 184 ft., as in the preceding example, and at an efficiency of about 40 per cent.

Another displacement pump is the Latta-Martin, designed chiefly for raising large volumes of water under low heads; though it may be constructed for any desired air pressure and head.† It consists of a pair of submerged cylindrical tanks,

* This series system of tanks has been proposed by E. A. Rix, *Transactions of the Technical Society of the Pacific Coast*, Aug. 3d, 1900, p. 187.

† *Compressed Air Magazine*, Jan., 1907, p. 4332.

taking water through large disk valves in the bottom. On the tops of the tanks is placed the valve mechanism for distributing the air alternately into each side. This valve gear comprises a main and auxiliary valve, each thrown by a piston valve, similar to those of many single-cylinder steam pumps. The movements of the valves are caused by the oscillation of a pair of levers, from each of which is suspended a bucket filled with water and hanging in a housing contained within the main tank. When the pump is in operation, the bucket housings are alternately filled and emptied of water, so that the difference in effective weight of the buckets causes them to rise and fall.

The Harris, or return-air displacement pump, made by the Pneumatic Engineering Co., uses the compressed air with some degree of expansion. There are two tanks, either submerged or within suction distance of the sump, each connected by a pipe with the compressor. The water enters by siphon action, the inlet, as well as the discharge valves, being placed above the tanks. Instead of being exhausted into the atmosphere at each stroke, after doing its work, the compressed air is conducted back to the intake of the compressor and expands behind its piston. Therefore, the system is a closed one, the same air being used over and over, in a manner similar to the operation of the Cummings return-air plant. The water chambers fill and discharge alternately, the admission and discharge of the air being governed by an automatic switch-valve, connecting the two air pipes close to the compressor.

In starting, after the water in one of the tanks has been expelled, the switch reverses and places this tank in connection with the compressor intake. Then, while the second tank is being discharged, the compressed air exhausted from the first returns to the compressor and, acting expansively upon the intake side of the piston, reduces by so much the power required to drive the compressor. When the pressure in the first tank has fallen sufficiently (by being in communication with the compressor intake), it will again fill with water. Thus, the compressor transfers the same body of air from one tank to the other, additional air to

make up for leakage being supplied through an adjustable check valve in the intake pipe. This valve is set to open during the suction period, at a negative pressure a little greater than the pressure required to draw water into the tanks. The switch-valve is operated automatically; either by a device acting at the intervals required to complete a cycle in both tanks, or by an electric make-and-break mechanism, controlled by a pressure gauge on the air intake. In the first case it would consist of a piston valve, operated by a small air cylinder, compressed air being admitted alternately to each side of the piston in the latter through an auxiliary valve. The volume of air required for a given size of tank may be determined in terms of revolutions of the compressor.

The Harris pump has a high efficiency, say fifty-five to sixty per cent., and requires but little attention during its operation. It may be adopted for shaft pumping by installing it in several units, one above another, according to the total lift.

The Halsey pneumatic pump is also made by the Pneumatic Engineering Co. It has a single, submerged tank, with a simple, automatic valve-motion, operated by a float.

If a displacement pump be required to work in acid water, such as frequently occurs in mines containing pyritiferous ore, the pressure tanks may be lined with concrete and the other parts made of bronze; or the tanks may be replaced by excavations in the rock, adjacent to the shaft and lined with concrete or asphalt.

Air-lift Pump. This, like the displacement pump, is a revival of an old principle. Since 1888, in which year Dr. Julius Pohlé proposed its application for pumping and erected an experimental plant, the air-lift has attained considerable prominence. Thus far it has been employed chiefly for raising water from deep wells, as for water-supply plant, but is applicable to a limited extent also for pumping in shafts and for elevating finely divided pulpy material mixed with water, such as the slimes and sands of cyanide and concentration mills.

The pump consists essentially of two pipes: a large column or delivery pipe and a relatively small air pipe, connected with the

compressor receiver. A diagram of the typical form of the apparatus is shown in Fig. 193. The delivery pipe, open at both ends, is submerged in the water to a depth proportionate to, but always greater than, the height to which the water is to be raised. The compressed-air supply pipe passes down to a point near the bottom, and terminates in a nozzle, which, directed vertically upward by a return bend, is inserted in the lower open end or foot-piece of the delivery pipe. (Modifications of this arrangement are noted hereafter.)

In some respects the operation of the air-lift pump is the reverse in principle of the method of compressing air by the direct action of falling water. As the compressed air leaves the small pipe it expands and, if the discharge pipe is of small diameter, tends to form piston-like layers, which rise rapidly, alternating with masses of water. This is readily shown by experimenting with glass tubes. But if the discharge pipe be of large diameter, the air should be admitted through

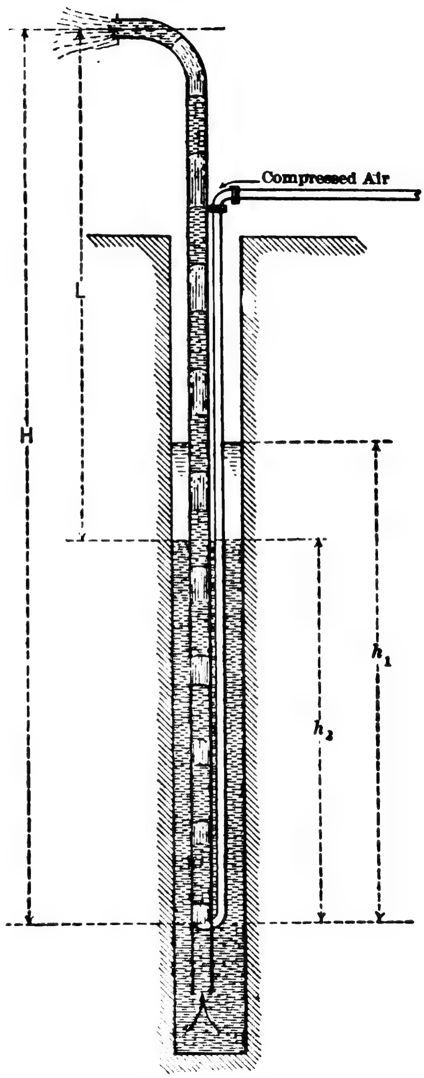


FIG. 193.—Diagram of Pohlé Air-Lift Pump.

a series of ports or nozzles, resulting in a dissemination through the rising water of small masses of air or bubbles. The water is raised chiefly by the buoyancy of the air; or, expressed differently, by the aëration of the column of water, which causes a reduction in its specific gravity. The action of the pump is due in a small degree only to the expansive force and vis viva of the compressed air. It is obvious that, before the air is turned on, the water stands at the same level inside and outside of the delivery pipe. On entering the foot-piece, the air is under a pressure due to the weight of the rising column of mixed air and water. As the bubbles of air rise, in forcing the water upward, they expand with the decrease in head; so that, on reaching the top of the column, the compression is that due only to the weight or pressure of the small quantity of water about to issue from the pipe. Thus, the air leaves the pump column at a pressure but little above atmospheric pressure. The initial air pressure required depends on the pressure due to head, measured from the nozzle or air ports to the surface of the water. If the pressure be too high loss of work ensues at the compressor. Should the delivery pipe be too deeply submerged, in proportion to the net height of lift, an uneconomically high pressure will be required to force the air into the foot-piece; and, with an insufficient submergence a larger quantity of air will be necessary to produce the velocity of delivery.

Referring to Fig. 193, let:

h_1 = depth to which the delivery-pipe foot-piece is sunk below the normal level of the water, before pumping begins, or when the water is at rest;

h_2 = height at which the water stands when the pump is in operation;

H = height of the column of mixed air and water, measured from the air inlet to the point of discharge;

L = net height of lift = $H - h_2$.

The compressed air enters the foot-piece at a pressure, P′, corresponding to the head, h_2; or, $h_2 \times$ pressure per foot of hydraulic head $= 0.434\ h_2$. Assuming that the water rises in piston-like masses—as would be the case with a single air nozzle and a

delivery pipe of small diameter—the sum of the lengths of these masses in the column H must be theoretically equal to the outside solid column of water, h_2. (The weight of the compressed air contained in the column may be neglected.) But, to overcome the frictional resistance and produce flow, the head h_2 must be greater. Under ordinary working conditions, the net height of lift, L, is found to be from 0.5 h_2 to say 0.65 h_2. Taking the second value and transposing: $h_2 = \frac{L}{0.65}$; and by substituting in the expression for the value of P′, as above: $P' = 0.434 \frac{L}{0.65} = 0.67$ L. If, for example, L be 50 ft., P′ = 33.5 lbs., and $h_2 = \frac{50}{0.65} = 77$ ft.

Since the air in the column H is divided into small masses, surrounded by water, its expansion during the upward flow may be assumed to be isothermal. If P′ be its initial pressure, the mean pressure for the entire lift $= P \times \text{Nap. log.} \left(\frac{P'}{P}\right)$, P and P′ being absolute pressures. In the above example, taking P as 15 lbs., P′ = 33.5 + 15 = 48.5 lbs., whence, the mean pressure = 17.5 lbs. gauge.

For starting the pump, the air pressure must be sufficient to overcome the normal static head, h_1, but, when the flow has begun, the pressure required falls to that corresponding to h_2. Though this difference in pressure (h_1—h_2) may be considerable, it is readily met by temporarily speeding up the compressor. To minimize fluctuations between h_1 and h_2, the top of the well or sump should be extended laterally, in order to furnish a large horizontal area of water, the level of which would be but little affected by stoppages or by variations in air pressure and delivery. The throttle valve in the air pipe may be regulated by a float on the surface of the water. Care should be taken in the design of the foot-piece and in properly proportioning the air pressure to the submergence and net lift. Otherwise, air may leak back into the sump or outside column of water; and, if this becomes aërated,

much more power and a larger volume of air will be required to keep the pump in operation. In such case the efficiency is greatly decreased.

Since 1889 many experiments by competent engineers have been made on the air-lift pump. Among the first were those of B. M. Randall and H. C. Behr, on a sixty-foot well, with a stage compressor. A summary of these tests is given by E. A. Rix, in the *Transactions of the Technical Society of the Pacific Coast*, Aug. 3d, 1900, p. 206. In 1894 a series of tests were made at De Kalb, Ill.,* and in 1893 and again in 1896 on four pumps at Rockford, Ill.†

The last-named were carefully carried out and the results compared in tabulated form. The heights of lift above water-level were 66.5, 90, and 91.5 ft., the air pressure being 76 lbs. gauge and the submersion, 225 ft. Both air pressure and depth of submersion appear to have been unnecessarily great. With a compressor of 124-h.-p., the net work done was 24-h.-p., or an efficiency of about 20 per cent. With 600 cu. ft. free air per minute, 200 cu. ft. of water were pumped, or 3 air to 1 water. The sizes of piping used were: delivery pipes, 4 in., 5 in., and 6¼ in., with air pipes from 1½ to 2½ in. In several of these tests the air pipe terminated in a ⅝-inch nozzle. The plan was also tried of closing the lower end of the air pipe and discharging the air through slot-shaped perforations in the sides near the bottom; but the results were inferior to those obtained from the single-nozzle opening. Possibly better work would have been done by some different arrangement or size of slots; for large pipes and volumes of water, at least, the single nozzle has not been found satisfactory.

E. E. Johnson gives a table of the performance of the air-lift pump, including consumption of power and theoretical and total efficiencies for different height of lift,‡ from which Table LV is abstracted:

* *Engineering News*, July 12th, 1894.
† *Ibid.*, March 4th, 1897.
‡ *Ibid.*, April 22d, 1897.

TABLE LV

Lift In.		Theoretical Horse-Power.				Efficiency of Air-Lift.				
		To Lift One Cubic Foot of Water per Minute.	To Deliver One Cubic Foot of Air per Minute.			Theoretical			Total Efficiency from Power Applied to Water Del'd.	
Pounds Pressure.	Feet Head.		Isothermal.	Two-Stage.	Adiabatic.	Isothermal Compression.	Two-Stage Compression.	Single-Stage or Adiabatic Compression.	Two-Stage Compression.	Single Stage or Adiabatic Compression.
5	11.54	.02185	.02514	.02572	.0263	.87	.848	.83	.623	.497
10	23.09	.04363	.05586	.05992	.064	.78	.728	.684	.546	.41
15	34.63	.06545	.09105	.0962	.1015	.72	.687	.648	.515	.389
20	46.20	.08727	.12994	.1391	.1483	.675	.627	.59	.47	.354
25	57.75	.109	.17191	.1897	.2004	.635	.575	.545	.432	.327
30	69.31	.13091	.21678	.2370	.2573	.603	.548	.508	.412	.305
35	80.86	.1527	.26445	.2915	.3187	.577	.52	.478	.39	.287
40	92.41	.17454	.31375	.3489	.3842	.557	.502	.455	.376	.273
45	103.90	.1963	.36368	.4085	.4535	.540	.482	.433	.362	.260
50	115.50	.21818	.41848	.4722	.5261	.522	.464	.415	.348	.249
55	127.00	.24	.47112	.5366	.6023	.51	.447	.40	.336	.24
60	138.60	.26181	.52855	.6051	.6818	.495	.432	.384	.324	.231
65	150.10	.2836	.58612	.6734	.7608	.483	.422	.372	.316	.223
70	161.70	.30545	.64812	.748	.8483	.471	.408	.36	.307	.216
75	173.30	.3273	.70952	.823	.9380	.462	.398	.35	.299	.210
80	184.80	.3491	.76843	.898	1.0291	.455	.39	.343	.292	.206
85	196.30	.37	.83039	.976	1.1231	.446	.38	.331	.285	.198
90	207.90	.3927	.89444	1.055	1.2176	.439	.373	.324	.28	.194
95	219.40	.4145	.96164	1.137	1.3148	.431	.368	.315	.276	.189
100	230.90	.43636	1.0243	1.247	1.4171	.428	.352	.308	.264	.185
110	254.10	.48	1.162	1.394	1.626	.413	.346	.296	.26	.177
120	277.20	.5236	1.301	1.571	1.841	.402	.333	.285	.25	.171
130	300.40	.5675	1.443	1.755	2.068	.394	.324	.275	.243	.165

These figures represent the work of well-proportioned plant, as to depth of submergence and air pressure. It is shown clearly that the efficiency of the air-lift falls off rapidly as the air pressure and height of lift increase. The higher efficiencies are naturally obtained from stage compression. In general it may be stated that, under normal conditions and with small lifts, efficiencies of from 30 to 35 per cent. are readily obtainable, and may rise to 45 or 50 per cent., with proper air pressures and ratios of submergence to height of lift.

In 1906 several tests were made at Wandsworth, England, on a modified Pohlé air-lift, with a delivery pipe of increasing diameter toward the top. The total height of the delivery pipe was 580 ft., of which 324 ft. were submerged, the net lift thus being 256 ft. In this case the distance $h_1—h_2$ was 69 ft., air pressure 135 lbs., ratio of volume of free air used to water discharged, 5.8 and 5.6 . 1. The total efficiency was 36 per cent. In view of the conditions this is an excellent showing and indicates an advantage in using a tapering column pipe.

The following results of a test made on a 300-ft. well will further illustrate this subject:*

Elevation of discharge above mouth of well.........	85 ft.
Depth to water-level during operation of pump	44 "
Net lift, water-level to point of discharge...........	129 "
Submergence of delivery pipe......................	248 "
Air admitted to delivery pipe 5 ft. above inlet end.	
Diameter of delivery pipe.........................	3.5 ins.
" " air pipe...............................	1.25 "
Volume of water delivered per minute..............	82.5 gals.
" " free air used per minute.................	81.8 cu. ft.
Gauge pressure of air.............................	107 lbs.
Consumption of free air per cu. ft. of water.........	7.44 cu. ft.
Horse-power consumed by compressor.............	12.1 " "
Total efficiency..................................	22.3 %

A number of calculated values for air-lift pumps are included in Table LVI.

The question of relative sizes of air and delivery pipes has not yet been satisfactorily answered. While there are many variations in practice, it is probable that ratios of diameter ranging from 1 : 2 up to 1 : 2½ or 3 will be found suitable. The absolute diameters of the pipes are determined on the basis of frictional loss caused by the flow of the air and water. A water velocity of 250 to 300 feet per minute may be assigned for the delivery pipe. The friction losses in air pipes have been discussed in Chapter XVI. It should be added that when the water is delivered at a

* G. C. H. Friedrich, *Trans. Ohio Soc. of Mech., Elec., and Steam Engrs.*, 1906.

distance from the pump, the additional frictional resistance must be determined, and the air pressure and submergence correspondingly increased. Reference may be made in this connection to a paper by H. T. Abrams, in *Compressed Air Magazine*, Aug., 1906, p. 4135.

TABLE LVI

Lift, Feet.	Volume of Air per Cubic Foot of Water.	Submergence, at Sixty per Cent. of Total Height of Delivery Pipe.	Air Pressure.	Horse-Power per Gallon Water per Minute.
25	2	38	17	0.0184
50	3	75	33	0.0426
75	4.5	113	49	0.0828
100	6	150	65	0.1320
125	7.5	188	82	0.1910
150	9	225	98	0.2544
175	10.5	263	115	0.3150
200	12	300	130	0.3808

Among the most complete and valuable recent tests of the air-lift pump are those made in 1907 by Messrs. Henderson and Wilson at the two 200-stamp mills of the Angelo and Cason mines, of the East Rand Proprietary Mines, Limited, South Africa.* At these mills both slimes and sands are raised to the settling tanks by air-lift pumps, instead of the usual tailings-pumps and wheels. The delivery pipes used in the 19 tests recorded were of two kinds, *viz:* 10- to 16-in. pipes of constant diameter, and several pipes increasing in diameter from 12 and 14 ins. at the bottom to 14 and 16 ins. at the top. These pipes did not taper uniformly, as this is impracticable; but, for a length of 35 ft. above the air inlet, were lined with one inch of wood, which served incidentally to protect the metal from the scouring action of the mixture of sands or slimes and water.

The foot-piece used in the earlier tests was flared out and closed at the bottom, the water and pulp being admitted through 4 large ports, 2½ ft. below the air inlet and having a combined area of about 200 sq. ins. The air inlet was a single opening, 4

* *The Engineer* (London), Jan. 10th, 1908, p. 26.

ins. diameter. For the later tests, the foot-piece was open at the bottom and modified by flaring it out to double the diameter of the column pipe, so as to increase gradually the velocity of inflow. And, instead of a single air inlet, a ring of twelve holes, one

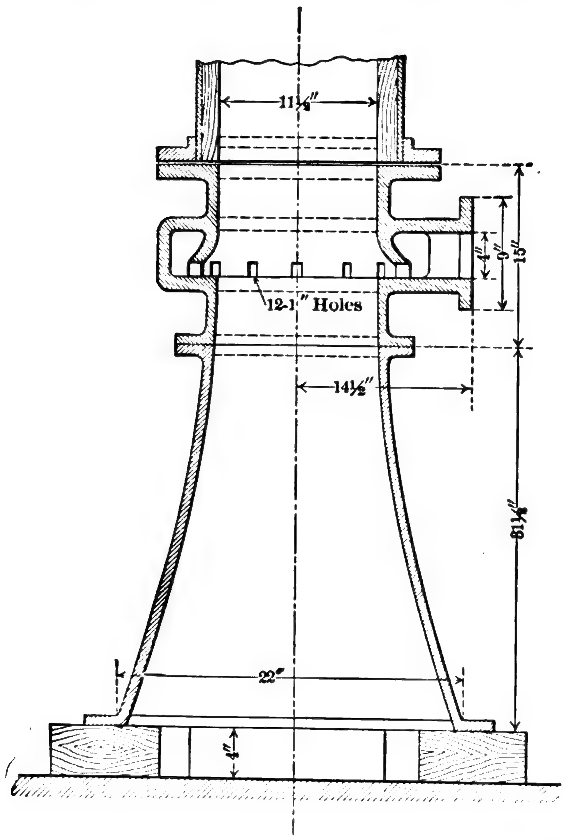


FIG. 194.—Foot-Piece for Air-Lift Pump, for Raising Mill Tailings and Slimes.

inch square, admitted the air; these holes being cast in an annular recess a little larger in diameter than the throat of the foot-piece.

The details of the modified foot-piece are given in Fig. 194. It is supported on timbers in such a way that the entire bottom

is open for the free admission of the material to be pumped. The column pipe is of steel tubing, expanded into cast-iron flanges, and lined in the lower part with wood, as already stated. This design gave materially higher efficiencies than the one first used, as set forth in the following table, which, though presenting the details of only four of the seventeen tests made, indicates the general results obtained. These results show that the air-lift, when properly designed for stated conditions, is sufficiently efficient to compete successfully with the tailings wheel, in common use in the district, and that it is superior to the tailings pump.

TABLE LVII

	Test.	1	2	3	4
Conditions.	Number and size of delivery pipes	Two 10-in.	Two 10-in.	One 16-in., decreasing to 14 ins.	One 14-in., decreasing to 12 ins.
	Submersion in feet	32.75	35.75	37.75	48.85
	Lift in feet	32.5	29.5	27.5	27.09
	Ratio of submersion to lift	1.009 to 1	1.21 to 1	1.372 to 1	1.77 to 1
	Gauge pressure of air, lbs.	15	16	17	22
	Kind of foot-piece	Original	Original	Modified	Modified
	Throat diameter of foot-piece	10 ins.	10 ins.	$13\frac{1}{2}$ ins.	$11\frac{1}{2}$ ins.
Performance.	Free air, cu. ft. per minute	2256	1279	746.48	846
	" " per cu. ft. of slimes	7.27	4.06	2.74	2.64
	Cu. ft. of slimes per minute	310	315	290	320
	Throat velocity, cu. ft. per second	4.7	4.8	4.85	7.39
	Theoretical horse-power in pulp raised	19.3	17.8	15.23	16.6
	Horse-power per cu. ft. free air compressed	.048	.050	.053	.064
	Air horse-power	108.72	64.74	42.21	54.14
	Efficiency, per cent	17.7	27.5	36.15	30.55

In the paper from which the above data are abstracted full details of all the tests are given. The conditions were modified in the progressive tests, as to the ratio of submersion to lift, diameter of delivery pipe, and air pressure. As a basis for calculating the theoretical horse-power represented by the mixture of water and pulp raised, the weight of the slimes was determined

to be 63.3 lbs., and of the sands, 64.56 lbs., per cu. ft. Thus, for the sands, this horse-power was taken to be:

$$\frac{(\text{Quantity of sands}+\text{water})\times 64.56\times \text{ft. lift}}{33{,}000}=.001956\times Q\times \text{ft. lift.}$$

The term "sands" refers to the mixture of water and ore as crushed by the stamps, from which the "slimes" have been separated in the milling process.

Lansell's Air-lift. An interesting modification of the air-lift pump, as applied by Mr. George Lansell to pumping water from a deep mine shaft in the well-known Bendigo district, of Victoria, Australia, may be described here. In the shaft in question water has been raised in a series of lifts from a depth of 1,385 feet. Fig. 195 shows diagrammatically the arrangement of the parts for two of the lifts.

The compressed air is conveyed from the receiver in a pipe, A, running down the shaft. The water is conducted from the tank or sump through a pipe, D, which first passes down the shaft a certain distance, depending upon the height to which the water is to be raised, and is then connected with an enlarged section of pipe, E, at the foot of the column or delivery pipe, B. Thus, the piping for each lift has the form of an inverted siphon, through the longer leg of which the water is discharged. At the lowest point of the siphon a short branch pipe, C, enters from the air main, A, the end of this branch being directed vertically upward into the foot-piece, E. Before the compressed air is turned on the water stands at the same level in the pipes D and B. The effect of this arrangement is similar to that produced by submerging in the body of water to be raised the lower part of the delivery pipe, as in the Pohlé air-lift pump. Check valves are placed, as shown, in the pipes D and C, to prevent air or water from passing back into the air pipe or into the tank. A throttle valve is provided in the pipe C, for regulating the supply of air as required. The relative heights of the various parts are not fixed, the dimensions as shown on the sketch indicating substantially the proper depth of the inverted siphon below the tanks, and the corresponding height of lift; thus, from the tank at the 250-ft. level, the pipe D passes

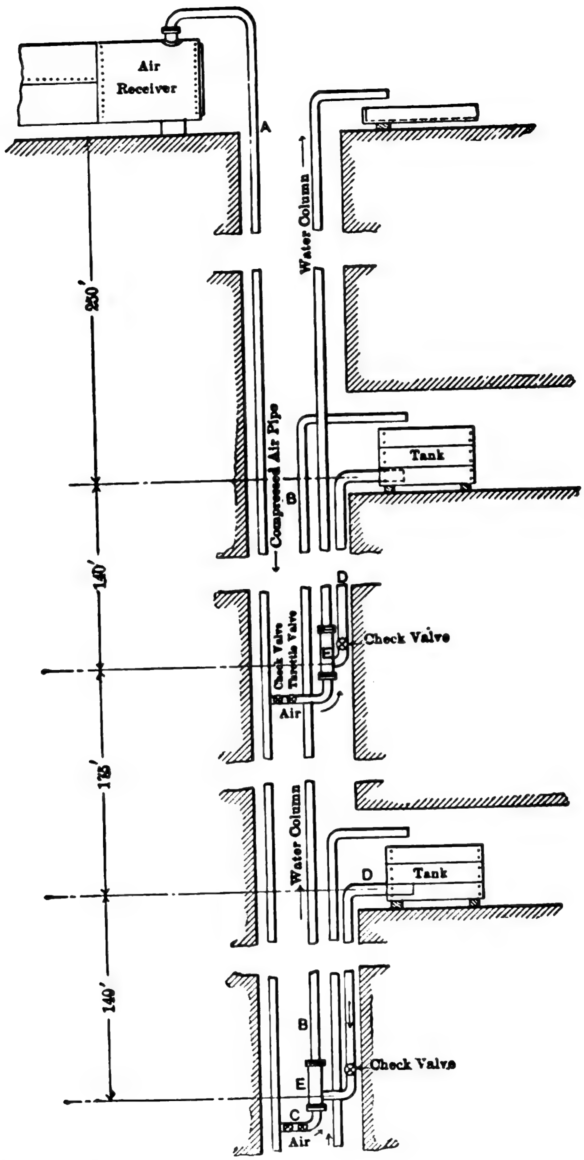


FIG. 195.—Diagram of Lansell's Air-Lift Pump for Mine Shafts.

down the shaft 140 ft., to the foot of the delivery pipe which discharges at the surface. A series of lifts may thus be arranged to raise the water from any desired depth. The pressure of air is the same for all, this pressure being sixty to eighty pounds per square inch, or that which is ordinarily furnished for mine service.

CHAPTER XXVI

COMPRESSED AIR HAULAGE

For the underground transportation of ore or coal, compressed air may be utilized either in locomotives or for driving stationary rope-haulage engines. Before taking up the subject in hand, a few considerations will be set forth respecting the operation of mine locomotives by steam and electricity as well as by compressed air. Steam locomotives are now much less frequently used than formerly for underground haulage, and they can be employed only in mines where the trains are conveyed through tunnels or entries directly to the surface, so that stoking may be done outside of the mine. Though uneconomical consumers of power, steam locomotives are rendered practicable in some collieries chiefly by the fact that the fuel is a product of the mines themselves and is therefore chargeable at a low cost. Their principal disadvantage lies in the serious vitiation of the mine atmosphere caused by the discharge into the workings of the products of combustion. Obviously they cannot be employed in gassy or fiery mines.

Electric and compressed-air locomotives divide between them a much broader field of operation. Both are applicable to mines of all kinds, whether collieries or metalliferous mines; for either long or short hauls, from a few hundred feet to several miles; they may be used underground in mines worked through shafts, where cars cannot be hauled through a tunnel to the surface, but must be hoisted on cages, and they do not vitiate the mine atmosphere. For underground haulage in mines containing firedamp, however, electric locomotives must be adopted with caution. Although, by the improvements introduced in recent years, much has been done to prevent the occurrence of serious sparking,

some risk from this cause still exists; and, furthermore, the possibility of strong sparking, accompanied by the momentary development of intense heat, from short-circuiting or by reason of a ruptured conductor, can hardly be averted.

Compressed-air locomotives were probably first used in the works of the Plymouth Cordage Co., Plymouth, Mass., about the year 1873; and in Great Britain, for mine haulage, in 1878, though these early designs were quite different from those now employed, and not very successful. Their introduction in the United States proceeded very slowly for some years. Perhaps twenty compressed-air locomotives were built previous to 1898, but since then they have been applied widely for a variety of service.* Expressed in general terms, the plant consists of a compressor (usually three-stage), receiver, pipe-line, charging stations, with the necessary valves and one or more locomotives. The storage tank or tanks carried by the locomotive are charged with a sufficient volume of high-pressure air for a round-trip run of the maximum length required, after which the locomotive returns to the nearest charging station for a fresh supply of air.

The special advantages of compressed air, as compared with electric haulage for mines, are: *First*, it may be used in collieries with perfect safety, in an atmosphere charged with fire-damp or dust, or in dry and heavily timbered workings; *second*, since the power is stored in the locomotive itself, the system presents the maximum degree of flexibility. The locomotives can enter all parts of the mine, wherever track is laid, far beyond the limit of the supply-pipe line, and are not, like electric locomotives, dependent upon wiring, which must accompany every foot of advance.† For collieries they may be used equally well for the haulage of trains on main lines, and for gathering and distributing individual cars among the working places; *third*, the compressed air costs little or nothing when not in actual use, and its

* Letter to the author from the H. K. Porter Co., Pittsburg, Pa.

† It should be noted, however, that storage battery and "cable-reel" electric locomotives have been introduced in a few cases, both in Europe and the United States. The latter has a very limited range of application and can be used for short branch lines only.

full power or but a fraction of it is available at all times. During the unavoidable periods of idleness of the locomotives no power is wasted, because, though the compressor may continue in operation at a slower speed, it is engaged in storing up power in the receiver and pipe-line. Incidentally the exhaust of the locomotives discharges fresh and cool air into the workings. While this is a minor consideration, it improves rather than injures the ventilation of the mine. Both electricity and compressed air must be looked upon merely as transmitters and distributers of power, depending for their production on either steam- or water-power as a prime mover.

At most mines compressed-air haulage is employed only for underground transportation, from the stopes or breasts to the foot of the hoisting shaft; in other cases, where the mine is worked through a tunnel or adit-level, the locomotives haul trains of cars direct to the breaker, tipple, or ore-bins, situated on the surface. Occasionally, as for example, at the Homestake Mine, Lead, S. D., compressed-air locomotives are used for surface transportation of ore, from the crusher houses at the shaft mouths to the different stamp mills; the object being chiefly to reduce the fire risk for the wooden structures, into and near which the haulage tracks pass. For the same reasons many plants have been installed in and about manufacturing establishments, containing inflammable buildings or materials, such as lumber yards and explosives factories or magazines.

Construction and Operation of the Locomotive. For mine service compressed-air locomotives have either one or two cylindrical storage tanks. These tanks, with the cylinders, piping, and other appurtenances, are mounted on a frame provided with springs similar to those of a steam locomotive and carried by 4 or 6 driving wheels. The 6-wheel type is used where a heavier locomotive or a lighter rail requires the distribution of the load over a greater number of points. Fig. 196 illustrates a recent design of a four-wheel, single-tank locomotive, as built by the H. K. Porter Co. It is made in several sizes, the details of which are given in the first four columns of the following table.

TABLE LVIII

Cylinders: diameter, inches	6	7	7	8	4	5
Cylinders: stroke, inches	10	12	14	14	7	8
Diameter of driving wheels, inches	23	24	24	26	18	20
Wheel-base usually desirable, feet and inches	3–0	4–0	4–0	4–0	1–10	2–9
Usual length over bumpers, feet	10 to 13	12 to 15	12 to 15	13 to 17	7–0	8–0
Usual length of tank, feet	7 to 9	9 to 12	9 to 12	10 to 14	3–6	4–6
Usual diameter of tank, inches	31½ to 40	33 to 40	33 to 40	33 to 42	30	30
Excess of width at cylinders over gauge of track, inches	24½	26	26	28	9	18
Extreme height, least desirable, feet and inches	4–8	5–0	5–3	5–6	4–9	4–6
Approximate cubic-feet capacity of tank	50 to 60	75 to 85	80 to 90	85 to 100	16 to 20	18 to 25
Maximum tank pressure usually desirable, pounds	800	800	800	800	800 to 900	800 to 900
Approximate weight in working order, pounds	10,000	15,000	17,000	20,000	5,000	7,000
Weight per yard of lightest rail advised, pounds	20	25	25	30	14	16
Radius of sharpest curve advised, feet	25	30	30	35	15	20
Radius of sharpest curve practicable, feet	15	16	16	20	10	12–6
Maximum pressure per square inch usually desirable for auxiliary reservoir, pounds	140	140	140	140	150	150
Tractive force, pounds	1,860	2,915	3,400	4,100	790	1,275

In the last two columns of the above table are details of the type of locomotive shown in side and rear-end elevation in Fig. 197. The one illustrated has 4×7-in. cylinders and is used on track of 27-in. gauge for hauling mine cars from the underground loading chutes to the shaft stations. The operator's seat is detachable, so that the locomotive can be readily transferred as required from one level to another, on a cage whose platform is 5 ft. long. These two sizes are suitable for general

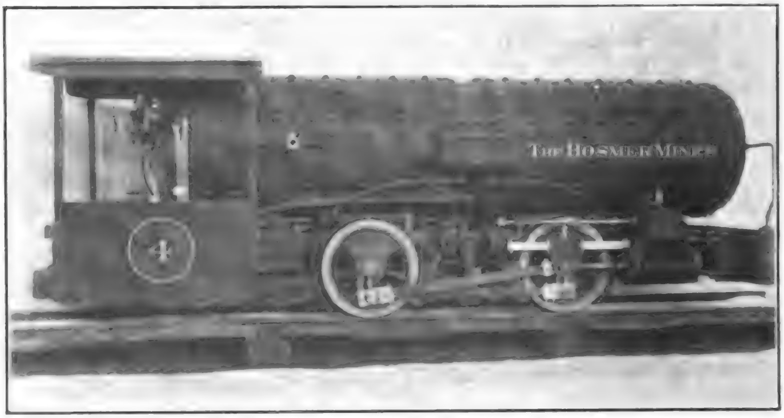


FIG. 196.

service in metal mines, or for gathering cars from individual working places in collieries, to make up trains on main haulageways.

A 6-wheel, double-tank locomotive, by the Baldwin Locomotive Works, is shown in Fig. 198. It has the following dimensions: gauge, 3 ft.; cylinders, 11 ins. $\times$ 14 ins.; main tanks, 22 ft. 7 ins. and 20 ft. 1 ins. $\times$ 34 ins. diameter, carrying a pressure of 800 lbs.; auxiliary tank pressure, 140 lbs.; driving wheels, 28 ins.; wheel-base, total, 6 ft. 6 ins.; total weight, 39,050 lbs., all on driving wheels. Another Baldwin locomotive, of the 4-wheel type, with 9×14-in. cylinders, 5-ft. 6-in. wheel-base, and weighing 24,350 lbs., is shown in Fig. 199. These builders make a number of other sizes of mine locomotive, the smallest weighing 8,000 lbs.,

FIG. 197.

and having $5\frac{1}{2} \times 10$-in. cylinders; track gauge, 36 ins.; tank pressure, 900 lbs., and working pressure 170 lbs. Some of the larger sizes are designed for a cylinder pressure of 200 lbs. Compressed

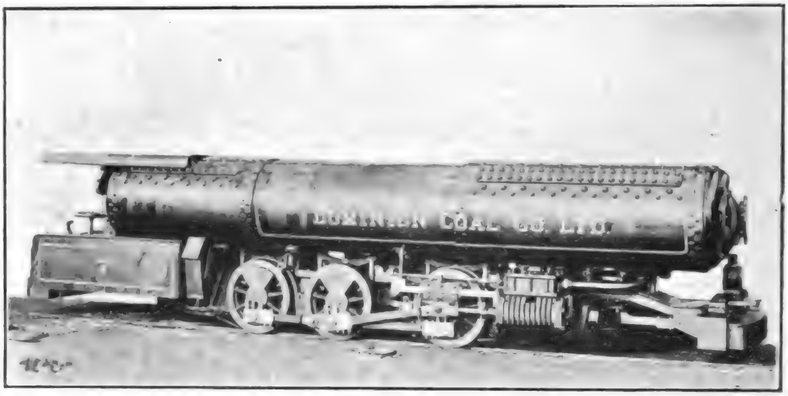


Fig. 198.

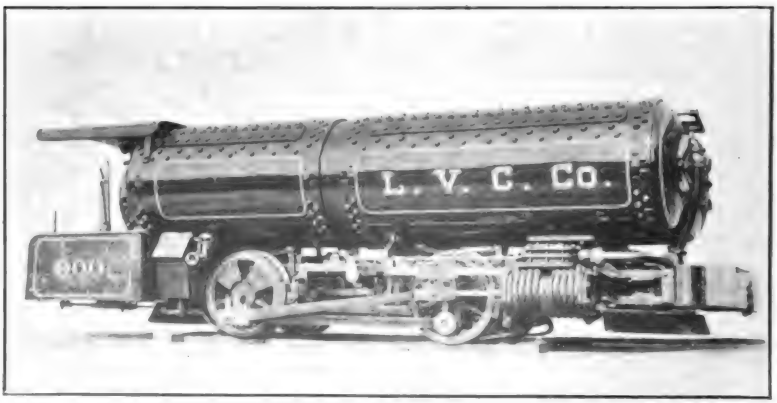


Fig. 199.

air mine locomotives are built also by the American Locomotive Company.

Where there are sharp curves in the track, as is commonly the case underground, the wheel-base must be short, say 4 ft. 6 ins. to 6 ft., for a 4-wheel engine. The height over all of the locomotive depends somewhat on the conditions existing in the mine

Table LIX

Cylinders { diameter, inches	7	8	9	10	11	12	13
Cylinders { stroke, inches	12	14	14	14	14	14	16
Diameter of driving wheels, inches	23	26	26	26	26	28	33
Wheel-base usually desirable, feet and inches	4–8	5–0	5–6	5–6	5–6	6–3	7–0
Usual length over bumpers, feet and inches	17–5	17–5	19–5	19–5	19–5	21–6	24–0
Usual length of tanks, feet	14 to 16	14 to 16	16 to 18	16 to 18	16 to 18	18 to 20	20 to 22
Usual diameter of tanks, inches	26½ to 28½	28½ to 31¾	31¾ to 34½	31¾ to 34½	31¾ to 38½	34½ to 38½	34½ to 42
Excess of width at cylinders over gauge of track, inches	26	28	30	32	34	38	42
Extreme height, least desirable, feet and inches	4–8	4–10½	5–3½	5–5	5–7	6–0	6–6
Approximate cubic-feet capacity of tanks	130	150	180	200	240	275	350
Maximum tank pressure usually desirable, pounds	800	800	800	800	800	800	800
Approximate weight in working order, pounds	18,000	23,000	27,000	31,000	37,000	43,000	51,000
Weight per yard of lightest rail advised, pounds	20	25	25	30	35	40	45
Radius of sharpest curve advised, feet	25	30	35	35	35	40	50
Radius of sharpest curve practicable, feet	20	20	25	25	25	35	40
Maximum pressure per square inch usually desirable for auxiliary reservoir, pounds	150	150	150	150	150	150	150
Tractive force, pounds	3,265	4,390	5,560	6,870	8,310	9,185	10,450

as to thickness of vein, head-room of the haulageways, etc., and is rarely more than 5 or 6 ft.—frequently less. The length varies greatly, mainly according to the tank capacity required, and the curvature of the gangways. It is usually from 10 to 15 ft. for the smaller sizes, up to 20 or 24 ft. for the larger, the widths ranging from $3\frac{1}{2}$ to 6 ft.

Table LIX contains the principal data of seven sizes of large six-wheel, double-tank locomotives, built by the H. K. Porter

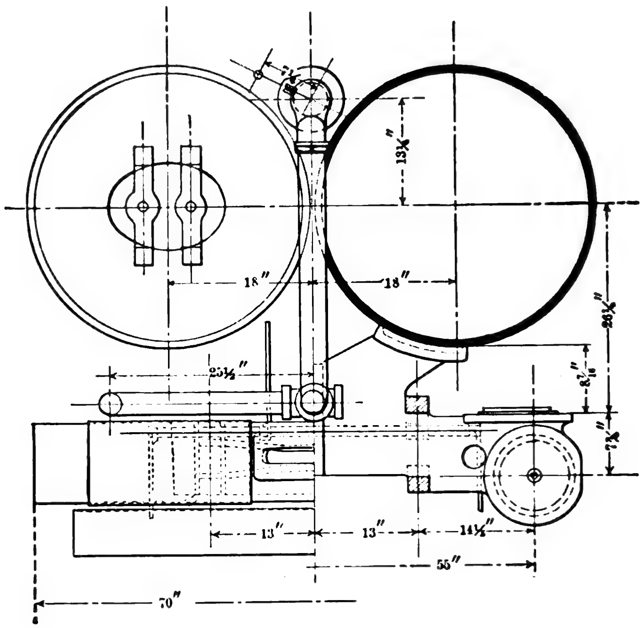


FIG. 201.

Co. These comprise the heaviest locomotives designed for underground mine service.

Additional details of the construction of compressed-air mine locomotives are exhibited in Fig. 200, containing a general plan and side elevation of a Baldwin locomotive, with 9×14-in. cylinders. Fig. 201 shows a half front-end elevation of the same, with half

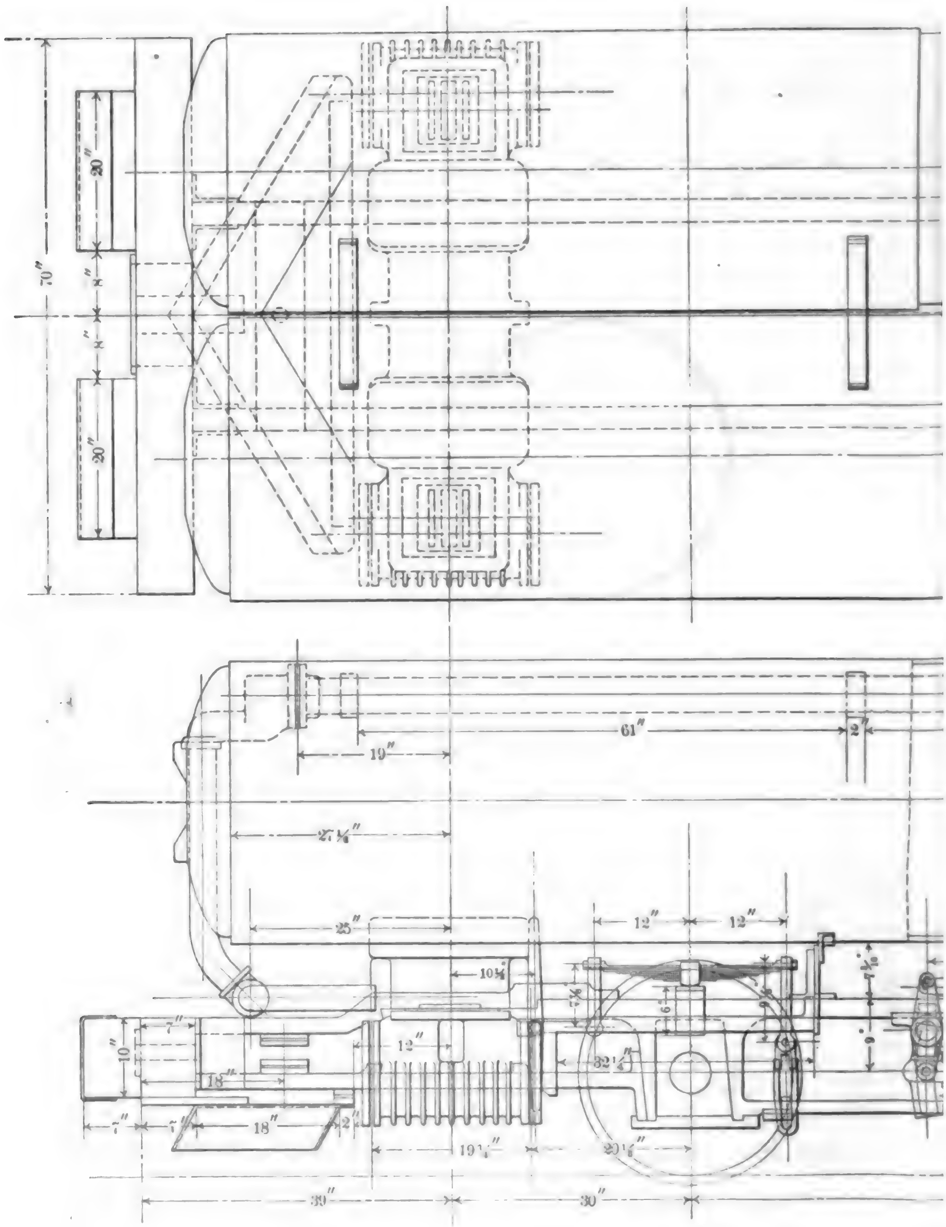
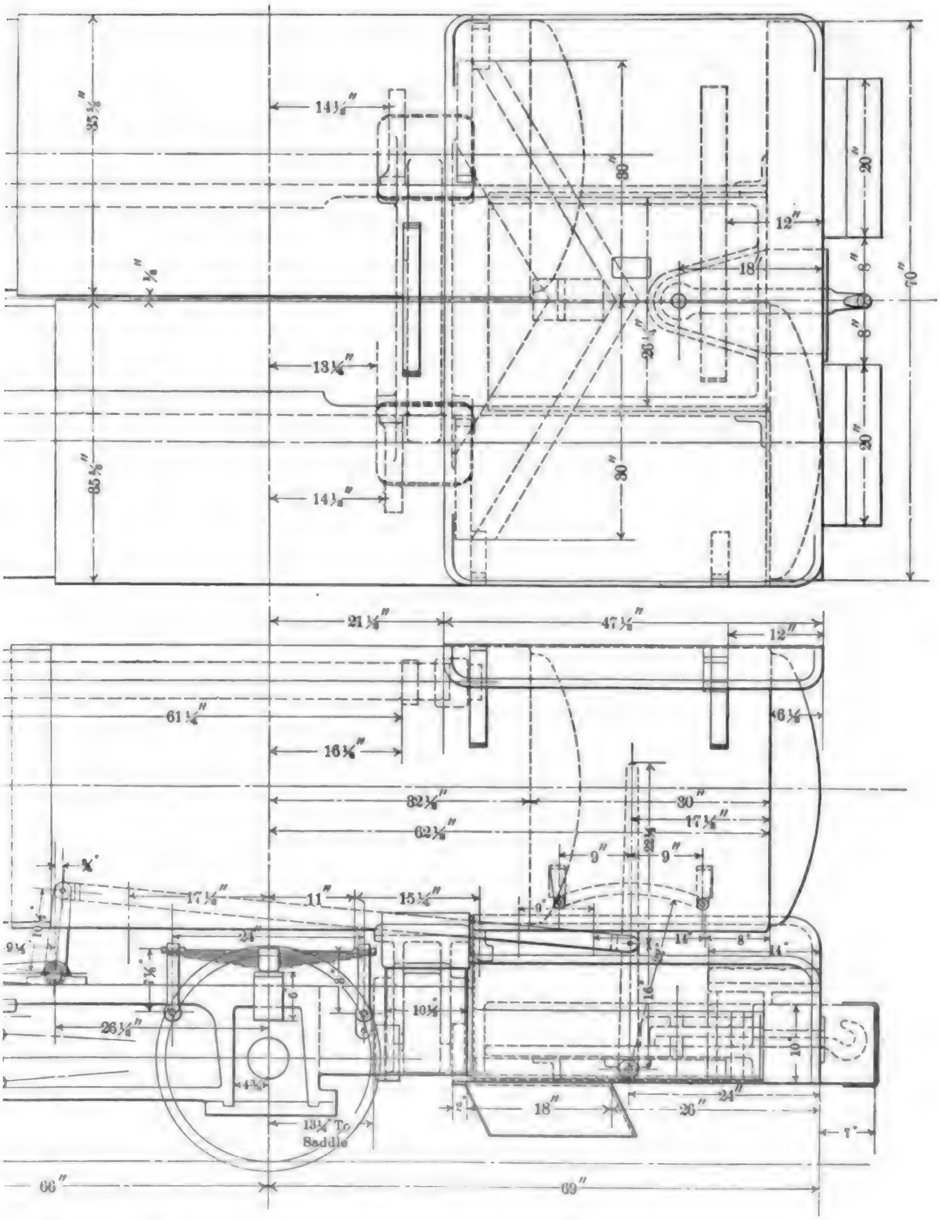


FIG. 200.—Baldwin Compressed-Air



Locomotive. (Plan and Elevation.)

section through frame and left-hand storage tank; and Fig. 202, a half rear-end elevation, with section of left-hand tank.

The tanks have dished or approximately hemispherical ends, and are built of extra heavy steel boiler plate; the shells being $\frac{3}{4}$ to $\frac{7}{8}$ in. thick, with 1 to $1\frac{1}{4}$-in. heads. Ring seams are double riveted with lap joints; longitudinal seams being butt joints, with inside and outside welt strips. As the tanks are generally built to carry working pressures of 700 to 800 lbs. per sq. in., the longi-

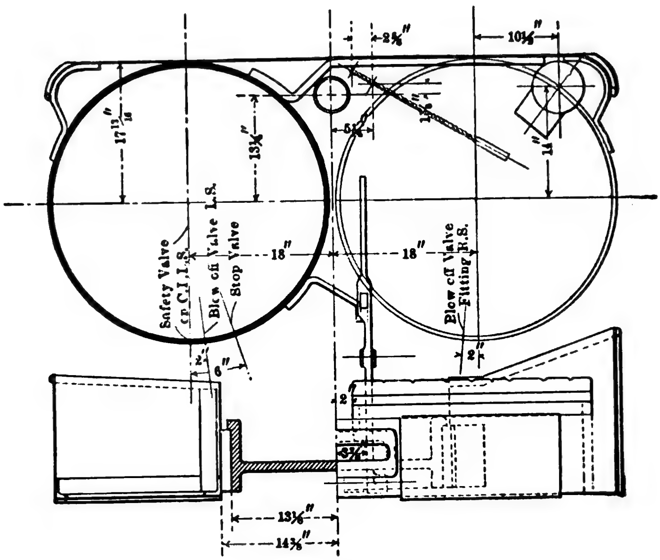


FIG. 202.

tudinal seams have 6 to 8 rows of rivets, to make a joint of not less than 75 per cent. of the strength of the plate. It is customary to test the tanks to 800 or 1,000 lbs., the factor of safety with plate of the usual quality being, say, $3\frac{1}{2}$. This is considered sufficient, as there are no strains produced by expansion and contraction, as in a boiler. When extremely high pressures are required, tanks of large diameter cannot safely be employed, and are replaced by a set of heavy seamless steel tubes, 8 to 9 ins. diameter—for example, the Mannesmann tubes. Tubes of this kind, 9 ins. diameter by

$\frac{11}{32}$ in. thick, will carry working pressures of 2,000 to 2,500 lbs. per sq. in. A number of them are laid together, bound by belts or straps and then enclosed in a light sheet-iron shell, to protect them from wet and rust. But these high pressures are unnecessary for ordinary systems of mine haulage.

From the main tanks the air passes into a small auxiliary or distributing reservoir and thence to the cylinders. This auxiliary tank is merely a section of wrought-iron pipe from 4 to 9 ins. diameter and 6 to 15 ft. long, with closed ends and laid alongside the main tank. By means of an automatic reducing valve, the pressure in the small reservoir is adjusted to the requirements of the engine. As used on the locomotives of the H. K. Porter Co., the reducing valve consists of a double-seated balanced valve, operated by a small piston. The air pressure in the auxiliary reservoir acts on one side of this piston and tends to close the valve. This action is opposed by a powerful external spring, which is adjusted to keep the valve open until the normal working pressure is reached in the auxiliary reservoir. Then the valve is closed by the air pressure, against the resistance of the spring. To provide for the case when the locomotive is using no air (as on a down grade or when at rest), a single-seated supplementary valve is placed in the pipe between the reducing valve and the locomotive storage tanks. This valve is controlled by the throttle lever; being open when the throttle is open, otherwise closed by the air pressure. By thus using two valves leakage from main tanks to auxiliary reservoir is avoided and a close regulation secured.

The cylinder pressure adopted ranges generally from 125 to 150 lbs., according to the size of cylinder and power required, thus being about one-quarter of the pressure in the main tank. From the small tank the air passes to the cylinders through a balanced throttle valve. This arrangement permits the maintenance of a constant working pressure, suited to the needs of the locomotive, prevents the waste of air likely to ensue if air at full tank pressure were admitted to the cylinders, and makes the locomotive more manageable. The cylinders, moreover, need not be made so

heavy as would be required for a high pressure. In starting a heavy load excessive slipping of the drivers is avoided, and with light loads the reducing valve may readily and quickly be regulated to produce any desired reduction of pressure. In the operation of the locomotive toward the end of the haul, when the pressure in the main tanks falls to that in the auxiliary tank, the cylinders take their air directly from the former, and the locomotive will continue to run as long as the pressure remains sufficient. Sometimes, for long hauls, and when the cross-sectional dimensions or sharp curves, or both, of the haulage-way do not permit the use of tanks of great length or large diameter, a tender carrying a supplementary tank is employed.

For small-scale work, the air is sometimes admitted to the cylinders throughout nearly full stroke, and consequently, as the exhaust is at high pressure, the efficiency is lower than it should be. This practice is doubtless due to the tendency to use as small a motor as possible for the service required, on account of the limited head room and narrow, crooked gangways so common in mines. Better results are obtained by using a cut-off and increasing the size of the locomotive and the weight on the drivers. This is almost always done with large locomotives. Ample reserve power is available when necessary, since full tank pressure can be temporarily admitted to the valve-chests in starting a heavy load, or in hauling on steep grades and around sharp curves. In using the air expansively, as can be done with properly proportioned cylinders, there should be no trouble from freezing of the moisture. Although the expansion will produce a low cylinder temperature, yet, as the initial working pressure is so much higher than is employed for pumps or other compressed-air machinery, the expanded air becomes relatively dry, and the force of the exhaust is still sufficient to keep the ports clear of accumulated ice. To this end the ports should be large, straight, and short, though ports of ordinary proportions are quite common. If high-pressure air were used in the engines, both cylinders and pistons would have to be made excessively heavy, and any reasonable degree of expansion would produce a degree of cold difficult

to deal with. The cylinders should not be lagged with non-conducting covering, as is so necessary for steam cylinders, to minimize condensation. By exposing their surface to the warm air of the mine, some heat is absorbed. Usually the exterior surface of the cylinders is cast with deep corrugations, in order to present the largest possible superficial area to the warm surrounding air. The cylinders are provided with slide valves; piston valves, like those used in steam locomotives, would leak more because of the dryness of the air.

On account of the cold produced by the reduction of pressure from the main tanks to the auxiliary reservoir, and to increase efficiency of operation, reheating is found to be advantageous, though not essential. It may be accomplished conveniently by applying heat to the auxiliary reservoir. If steam be available in the mine, a quantity of steam and hot water may be injected into this reservoir each time the locomotive is charged. Or, in mines where there is no danger from fire-damp, a small reheating apparatus for burning oil or coke may be carried on the locomotive. It is always desirable to warm the reducing valve from the main tank, as this is subjected to intense cold. In any case, when the air is reheated a quantity of water should be kept in the small tank. An incidental advantage of this arrangement is that the moisture from the hot water, which passes with the air into the cylinders, assists in lubricating the valves and pistons.*

Pipe-line and Charging Stations. The capacity of the compressed-air system naturally depends on the length of haul and size of locomotives, as influenced by the daily output, weight of trains, and gradients of the haulage lines. For short hauls, the pipe-line is sometimes omitted altogether, the locomotive returning each time to the compressor receiver to be recharged. In general practice, however, a pipe-line is carried underground, and at one or more points charging stations are established. The location and distance apart of these stations is determined by the haulage distances and the storage capacity of the locomotive

* E. P. Lord, Paper Read before the Anthracite Coal Operators' Association, N. Y., Oct. 13th, 1897.

tanks. It is evident that the last or innermost charging station, farthest from the compressor, must be at a point from which the locomotive can reach the end of its trip and return for a fresh supply of compressed air. For very long hauls, heavy traffic, or adverse gradients, a charging station may be required at each end of the line.

It is unnecessary to provide receivers inside the mine, though this may be done advantageously if the diameter of the supply pipe is small. The pipe-line itself is intended to act as a storage reservoir, and should be of a diameter which, in proportion to its length, will furnish a cubic capacity sufficient to charge the locomotive tanks quickly and without serious drop in pressure. In other words, when the locomotive is connected with the pipe-line, and the charging valve opened, the pressure in the locomotive tank and in the pipe, on equalizing (as it must), should not fall much below the stated pressure which the locomotive is designed to carry. It is, therefore, desirable that the volume of storage, represented by the main—or main and receiver—should be at least three times the tank capacity of the locomotive. To determine the necessary storage capacity of pipe-line, or combined receiver and pipe-line, several variables must be harmonized, as follows:*

V = storage volume required, in cu. ft.
v = volume of locomotive tanks, in cu. ft.
P = pipe-line pressure, in lbs. per sq. in.
p = desired pressure in locomotive tanks, in lbs. per sq. in.
p' = residual pressure in locomotive tanks, just before charging, in lbs. per sq. in.

$$\text{Then: } V\,(P-p)=v\,(p-p'), \text{ or } V=\frac{v\,(p-p')}{P-p}$$

For example, let $P=900$ lbs., $p=750$ lbs., $p'=125$ lbs., and $v=100$ cu. ft., from which:

$$V=\frac{100\,(750-125)}{900-750}=416.6 \text{ cu. ft.}$$

By transposition, the same formula may be used for finding

* H. K. Porter Co., "Handbook of Compressed-Air Haulage," 1907.

the pipe-line pressure required to produce a given pressure in the locomotive tanks. When several locomotives are served by the same pipe-line and compressor it is rarely, if ever, necessary to design the system for charging more than one at a time. If the volumetric capacity of the pipe-line be ample, the relatively small drop in gauge pressure on charging is soon recovered by the compressor, which, except in plants operating a single locomotive, is kept in nearly constant operation. In case additional locomotives are required after the original installation of the system, the same pipe-line may still serve, provided the compressor be of sufficient size to charge it to full pressure at shorter intervals.

The piping, which generally varies in diameter between 3 and 5 ins.—sometimes 6 ins.—should be of the best material, lap-welded, and with sleeve joints made with the utmost care to prevent leakage. To stop leaks, the sleeves should have annular grooves at each end into which soft metal calking is driven if required. It is advisable not to bury the pipe alongside the track, but to carry it entirely uncovered along one side of the tunnel or gangway, either on the floor or on brackets, so that leaks will at once attract attention and be stopped. While an occasional bend in the pipe-line is advantageous in permitting free expansion and contraction, they should not be too numerous, as they involve more joints and therefore a greater possibility of leakage.

Charging Apparatus. A common form of apparatus for charging the locomotives, as shown in Fig. 203, consists of a vertical right-angled connection inserted in the air main by means of a heavy tee. This connection has an arm projecting from the main a sufficient distance for conveniently coupling to the charging pipe of the locomotive. It comprises two parts: a vertical, rigid branch, containing a strong, accurately fitted 1½-inch gate-valve, and a short horizontal pipe, attached to the valve by a union and a ball-and-socket or flexible joint, for coupling to the locomotive charging pipe. Thus, the locomotive need not be stopped at a precise point for charging, but has a foot or two lee-

way on its track. When not in use, the flexible connection is swung back, out of the way. In the locomotive connection there are usually two ball-and-socket joints, together with a check-valve close to the tank.

After coupling on the locomotive, the gate-valve is opened, whereupon the air pressure immediately forces together the parts of the ball-and-socket joints and makes a perfectly tight connec-

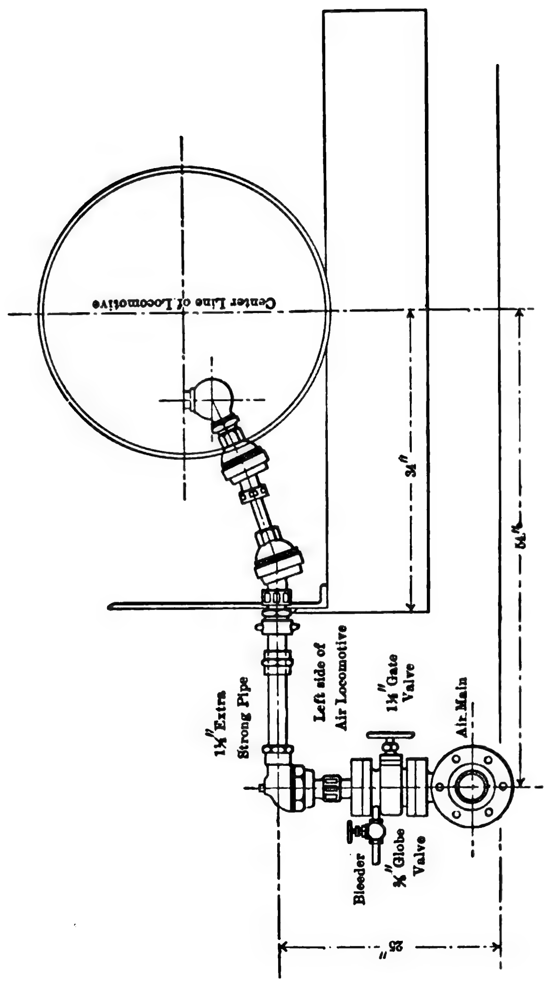


FIG. 203—Locomotive Charging Station.

tion. As soon as equilibrium is established between the pressures in the main and the locomotive tank the gate-valve is closed. To break the coupling, the compressed air remaining in the connecting pipe, between the gate-valve and the locomotive check-valve, must first be released. This is done by opening a small "bleeder valve," placed just above the gate-valve, as shown in the cut. The joints then become loose and are readily manipulated. The actual time occupied in charging is very short (usually about three-quarters of a minute), owing to the high pressure in the main and the relatively large diameter ($1\frac{1}{2}$ in.) of the charging pipe; but, including stopping the locomotive and making the connection, $1\frac{1}{2}$ to $2\frac{1}{2}$ mins. may be allowed. Frequently, charging may be done during the necessary delays in shifting cars and making up trains.

Calculation of Motive Power. To determine the motive power required for a given output, several factors must be known, *viz:* the tractive resistance per ton of the loaded cars on a level, the resistances due to gradients and curves, the weight of empty and of loaded cars, and the number of cars to be hauled in each train. The values of these factors are known approximately or are readily ascertained, with the exception of the resistances due to curvature of track and character of roadbed. The former has been determined experimentally for ordinary surface railways, but underground mine track is apt to be roughly laid, with curves of varying and irregular radius, and the elevation of the outer rail improperly adjusted. With sufficient weight on the drivers, however, sticking on a curve may be avoided, in the case of compressed-air haulage, by temporarily admitting to the cylinders a little air at full tank pressure, as already noted. In this respect compressed-air locomotives possess a material advantage over those driven by steam, in which the working pressure is limited and practically constant.

The average tractive force required per ton depends not only on the physical condition of the track and roadbed, but on the character and state of repair of the running gear of the cars. On level mine track the coefficient of rolling friction should usually

be taken at from thirty to forty pounds per ton, though it may be considerably higher on poorly laid or light track, or at the instant of starting the load. With mine track in exceptionally good condition, the coefficient may be as low as twenty pounds per ton. The grade resistance is twenty pounds per short ton, for each one per cent. of grade. Not infrequently, the distribution of grades on the haulage lines is such that the maximum load is not the resistance of the loaded trains, which are usually hauled on slight down grades, but that of the return trains of empty cars on the adverse gradients. To obtain the most economical results, gradients should be not over ½ to ¾ of 1 per cent. in favor of the loaded trains. With mine track and rolling stock of ordinary character, and a grade of 5 to 6 ins. per 100 ft., the coefficient of rolling friction is nearly the same for a loaded train hauled down as for an empty train of the same number of cars hauled up the grade. Heavier and even adverse grades often become necessary—sometimes as steep as 2½ per cent. to 3 per cent. or more, but they should be avoided as far as possible, because the maximum tractive force of the locomotive falls off rapidly. On a 2½-per-cent. adverse grade the locomotive can haul only about 4 times its own weight, even if the track be not slippery. Grades should be reduced on curves. Colliery cars, carrying 2½ to 3½ tons, will weigh from 1,800 to 2,300 lbs., while those used in metalliferous mines, where mechanical haulage is employed, vary between, say, 1,000 and 2,000 lbs. Many cars of the last-named weight are in use, for example, in the iron mines of the Northwest. Finally, having ascertained as near as possible the values of the different factors, the proper allowance of reserve power, in terms of volume and pressure of air, to cover indeterminate additional resistances due to imperfections of track and rolling stock, is a matter of judgment and experience.

With a given air pressure, the capacity required for the locomotive storage tanks depends primarily on the length of round trip to be made with a single charge of air. When this distance is, say, 1 to 1½ miles, the tank capacity generally varies between 50 and 150 cu. ft., according to the load; which, in turn, together

with the track and grade resistances, governs the dimensions of the cylinders. Cylinders of 5 ins. × 10 ins. up to 9 ins. × 14 ins. are commonly used for mine service, the larger sizes being adopted for heavy work in collieries. Still more powerful locomotives are used for some kinds of surface work. In several installations, as at mines of the Philadelphia & Reading Coal & Iron Co., the compressed-air locomotives have been designed with compound cylinders. For long runs, of over one and one-half miles, it is often desirable to increase the air pressure, rather than build tanks of very large size. Another plan is to provide a tender, which carries one or more auxiliary tanks, connected with those on the locomotive. Very long runs can be made by this means.

Having determined the total work in foot-pounds to be done with a single charge of air, on a run of the maximum length, specifications may be obtained from the builders for a locomotive of suitable weight, gauge, wheel-base, tank capacity, and cylinder dimensions.

Compressors for Charging Pneumatic Locomotives. For compressing the air to the high tension required by pneumatic locomotives, the work must be done in at least 3 stages; 4-stage compressors are sometimes employed for pressures exceeding 900 or 1,000 lbs. Efficient intercoolers are of course placed between the successive cylinders and an aftercooler is desirable. Fig. 204 shows the standard type of 3-stage locomotive charger built by the Norwalk Iron Works Co., for pressures up to 1,000 or 1,200 lbs. The air passes from the low-pressure cylinder to the lower of the two intercoolers and, thence to the intermediate cylinder. From the latter the air is delivered through the vertical pipe to the upper intercooler, whence it passes through the inclined pipe to the high-pressure cylinder. From this cylinder the compressed air is delivered to the receiver through the connection indicated under the outer end of the cylinder. Other compressors by the same builders are designed for pressures up to 3,000 and 4,000 lbs.

The air end of a three-stage, tandem locomotive charger, built by the Ingersoll-Rand Co., is shown in longitudinal section

in Fig. 205. The high-pressure intercooler is placed in the lower right-hand corner of the cut. Figs. 206 and 207 illustrate respectively the low- and high-pressure air ends of a duplex, four-stage compressor. In Fig. 206 are the intake and first intermediate cylinders, and in Fig. 207 the second intermediate and high-pressure cylinders. A perspective view of a large compressor of this type is shown by Fig. 208.

It will be seen in the sections that the pistons of the high-pressure cylinders are solid rams or plungers, provided with a series

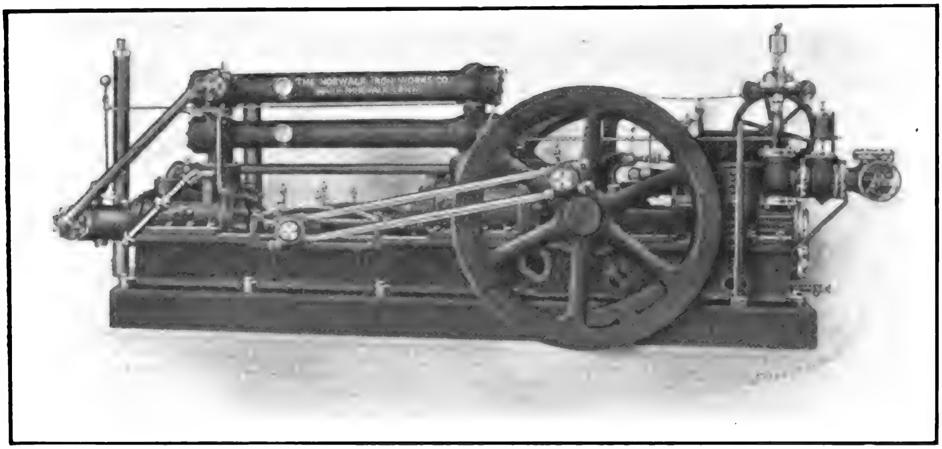


FIG. 204.—Norwalk Locomotive-Charging Compressor.

of packing rings. These, with the high-pressure valves, must be made with special care, to prevent the serious effects of leakage of high-pressure air. Even a small percentage of leakage will greatly reduce the volumetric capacity and efficiency. Locomotive chargers are also built by the Sullivan Machinery Co. and others.

When the mine is already provided with an ordinary low-pressure air plant, for machine drills and other service, and which has some surplus capacity, a two-stage charging compressor may be installed, to take air from the low-pressure system and bring it

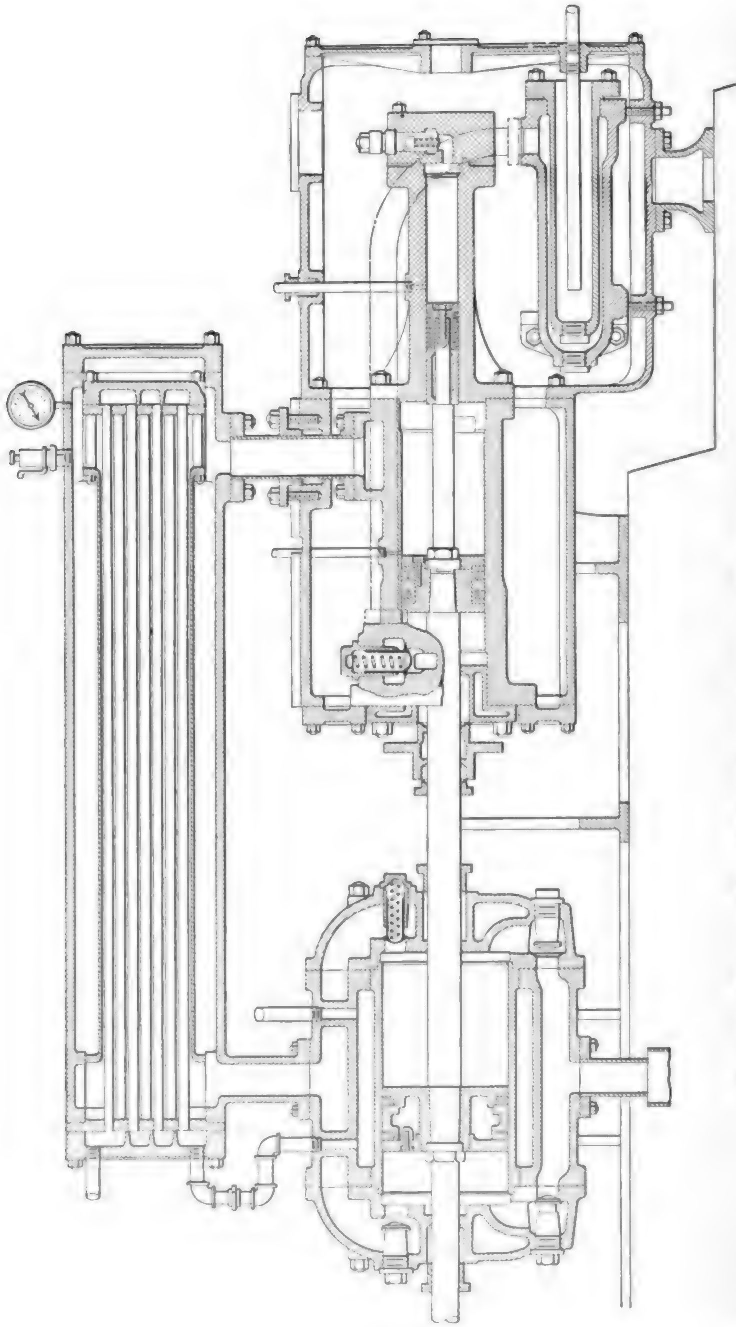
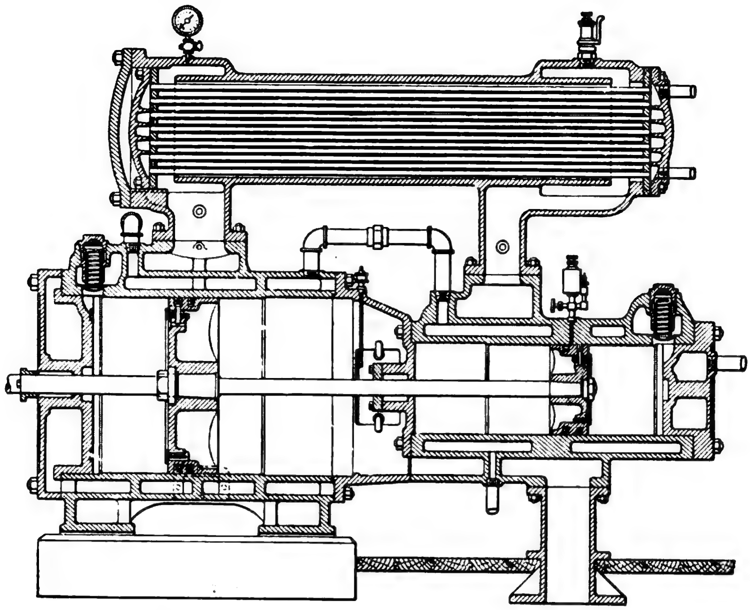
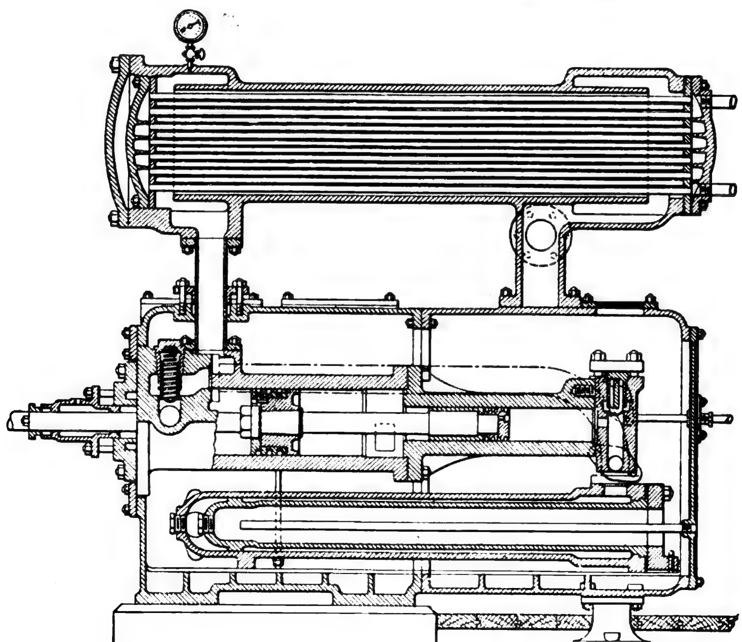


FIG. 205.—Ingersoll-Rand Three-Stage Locomotive Charger.





FIGS. 206 and 207.—Ingersoll-Rand Four-Stage Locomotive Charger.

up to the tension required for the locomotives. By this arrangement some reduction in the cost of the plant may be effected. Care must be exercised, however, in making such a combination. If the quantity of air produced by the low-pressure system should at times be insufficient to furnish the necessary excess, at ordinary gauge pressure, for the locomotive-charging compressor, the latter might be compelled to compress from too low an initial pressure.

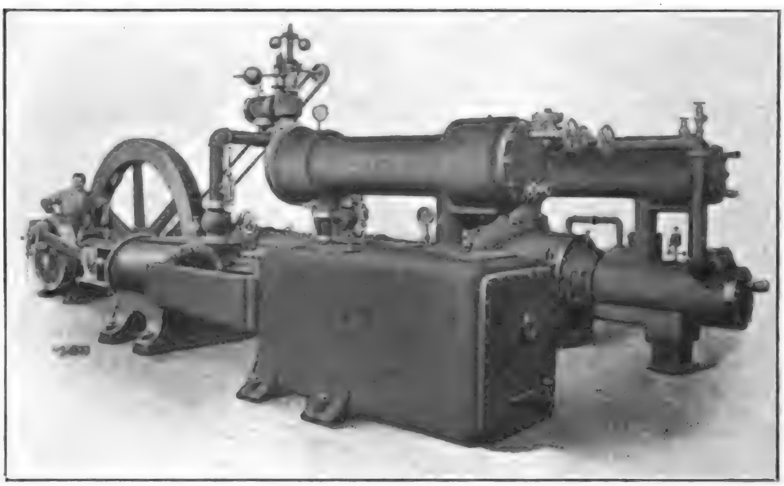


FIG. 208.

This would cause excessive development of heat and, aside from the difficulty of maintaining proper lubrication, might possibly raise the cylinder temperature to the flashing-point of the oil, thus causing an explosion. This matter has been discussed in Chapter XIV. Generally, it is preferable to install an independent locomotive charger. With such a compressor, the final temperature can be kept down to a moderate degree, say, 200° to 300° F., provided the plant is not too small for its work. The compressor should be run at a moderate speed, and as the demand upon it is usually somewhat irregular, causing frequent reductions of speed, and even occasional stoppages, the air cylinders are prevented from becoming over-heated.

The capacity of the charging compressor depends on the pipe-line pressure to be maintained, the number of locomotives to be operated, the cubic contents of the locomotive tanks, the pressure carried by the system, and the relation between the charging periods.

Let C = compressor capacity required, in cubic feet of free air per minute.

L = locomotive-tank capacity, in cubic feet of free air per minute.

N = number of charges required in any given time, T.

Hence the equation: $C = \frac{N\,L}{T}$

For example, if N = 3, L = 5,200 (corresponding to 100 cu. ft. of air at 750 lbs. gauge pressure), and T = 60 minutes:

$C = \frac{3 \times 5,200}{60} = 260$ cu. ft. free air per minute.

When the locomotives are charged—as they usually can be—at approximately equal intervals of time throughout the day, a single application of the above formula will be sufficient. Otherwise, calculations are required to determine the maximum and minimum rates of consumption of air. It is hardly necessary to add that, when the plant is installed at an altitude above sea-level, allowances must be made for decreased output, as explained in Chapter XIII.

Examples of Compressed-Air Haulage Plants. In further illustration of this subject, some of the details of a few successful installations may here be given.

1. At the Buck Mountain Colliery, Penn., are two 8-ton H. K. Porter locomotives, each with 2 tanks, respectively, 15 and 17 ft. long, having a combined capacity of 130 cu. ft. of air at 550 lbs. pressure.* The cylinders are 7 ins. × 14 ins.; wheel-base, 5 ft. 3 ins.; gauge of track, 42 ins.; height, 5 ft. 2 ins.; length over all, 19 ft. A round trip of 5,100 ft. is made in 30 to 40 minutes, or 2,500 ft. in 12 to 15 minutes, with trains of 12 cars, on grades of ½ to 4½ per cent., averaging ¾ of 1 per cent. in favor of the load.

* *Mines and Minerals*, July, 1898, p. 538.

One locomotive delivers 150 cars per 10 hours, doing the work formerly done by 15 mules. Weight of cars, 3,400 lbs. empty, and 10,400 lbs. loaded. A 3-stage Norwalk compressor supplies 375 cu. ft. free air per minute, at 700 lbs. gauge. Pipe-line, 4 ins. diameter and 9,600 ft. long, with a storage capacity of 800 cu. ft.

Average cost per ton-mile: 1.875 cents for the gross weight hauled, or 3.77 cents for net weight of coal. The cost for mule haulage under the same conditions was formerly 3.94 and 7.92 cents, respectively.

The cost of this plant was as follows:

Two locomotives		$5,505.
Air line: 9,647 ft. 4 in. pipe	$2,894.	
Six charging stations	360.	
Fittings and valves	382.	
Labor cost for erection	998.	
		4,634.
Compressor	$2,880.	
Sundries and erection	246.	
Compressor house	256.	
Steam line to compressor	152.	3,534.
Total cost		$13,673.

2. Empire Mine, Grass Valley, Cal. Several small compressed-air locomotives, built by Edward A. Rix, are employed in the deep levels of the mine, for hauling trains of 5 cars, each carrying 1 ton. The maximum distance covered by a round trip is about 5,000 ft. Locomotive storage tank measures 36 ins. diameter × 48 ins. long, carrying a pressure of 500 lbs. The dimensions over all are only 5 ft. long × 30 ins. wide × 52 ins. high, the gauge of track being 18 ins. One of these locomotives (Fig. 209) is operated by a pair of vertical engines, a chain and sprocket drive connecting the crank-shaft with the rear axle. There are 2 tandem tanks, one of them being carried on a tender. A reheater, provided with a Primus kerosene burner, reheats the air after its pressure has been reduced in the auxiliary reservoir. Mr. Rix has recently built 3 similar locomotives, but with a single, larger tank, for a 3-mile tunnel, near San Francisco. They carry

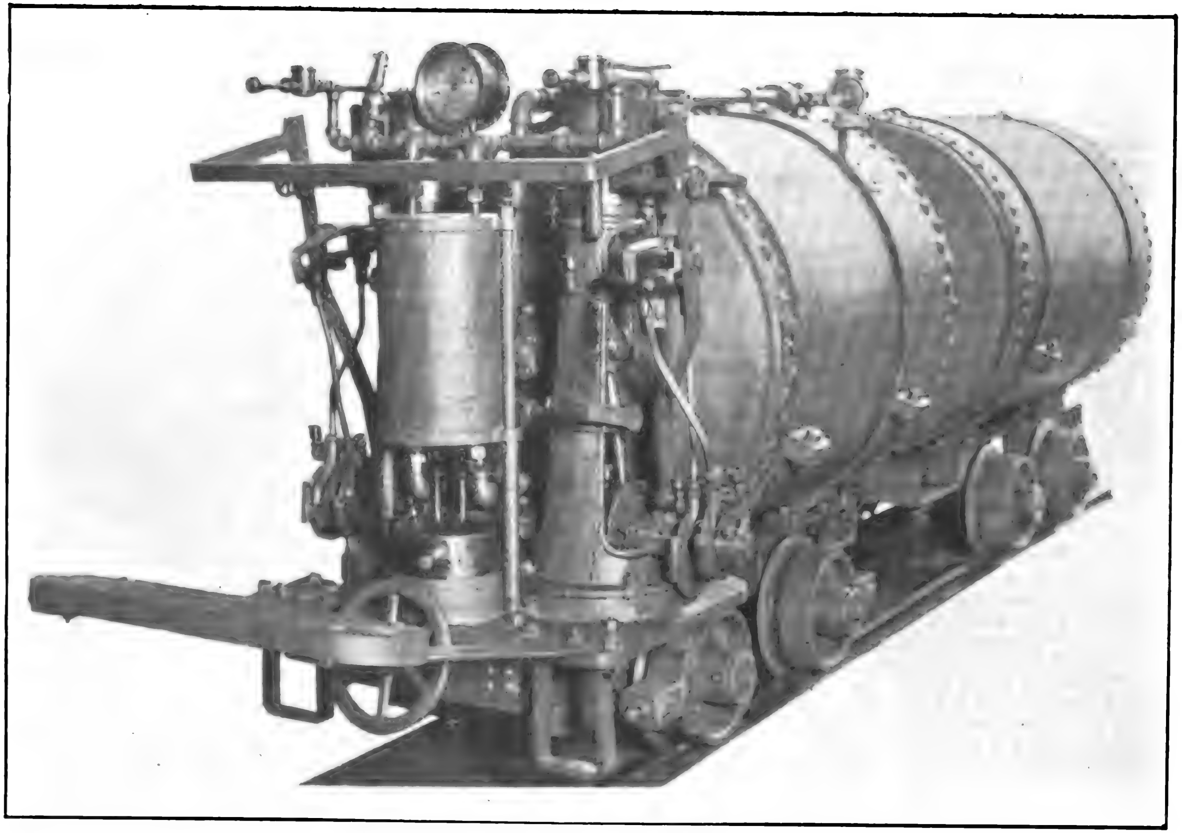


FIG. 209.

1,000 lbs. tank pressure, the working pressure being 100 lbs.; each locomotive making a 2-mile round trip, at 6 to 7 miles per hour.*

3. The Peerless Colliery, Vivian, West Va., operated for years several H. K. Porter locomotives, with 5×10-in. cylinders and weighing 10,000 lbs. Over all dimensions: 10 ft. 5½ in. long ×5 ft. 8 ins. wide × 4 ft. 5 ins. high. Four driving wheels, 23 ins. diameter; gauge, 44 ins. Capacity of main storage tank, 47 cu. ft.; pressure, 535 lbs.; charging time, 20 seconds; working pressure, 125 lbs. Pipe-line, 3 ins. diameter, with a total capacity of 242 cu. ft. Line pressure, 735 lbs. Trains consist of 6 cars, each weighing loaded 8,500 lbs. Grades range from level to 2½ per cent., generally in favor of the load. Curves from rooms to haulageways, 23 ft. radius, though locomotives are designed to work on curves as sharp as 15-ft. radius. Length of maximum round trip, 9,000 ft.; maximum speed 10 to 12 miles per hour. Cost of each locomotive, $1,800.

4. The following data, concerning one of the plants of the Philadelphia & Reading Coal & Iron Co., and compiled by Mr. G. Clemens, a division engineer of the Company, are published in the catalogue of the Baldwin Locomotive Works:

a. Shaft level—1 locomotive.

Round trip, 5,200 ft.; grades $\frac{4}{10}$ to $\frac{8}{10}$ of 1 per cent., all in favor of load; charging station at each end of run; gauge of track, 44 ins.; 40-lb. rails; weight of cars—empty, 3,300 lbs., loaded, 8,800 lbs.; from 15 to 38 cars per trip; total output, 600 cars per 10 hours. Round-trip time, 12 min.; charging time, 1 min. A round trip and a half can be made with one charging.

b. Slope level—1 locomotive.

Length of haul, 3,200 ft., of which 700 ft. is on a slope whose grade ranges from $4\frac{1}{10}$ to $5\frac{1}{2}$ per cent. Grade of main gangway, $\frac{4}{10}$ to $\frac{8}{10}$ of 1 per cent., in favor of load. Trains of 10 cars are hauled on main gangway, and 4 cars on the slope; weights of cars same as above.

Locomotive-tank pressure at start, 600 lbs.; at end of trip,

* *Compressed Air Magazine*, Feb., 1908, p. 4747.

200 lbs. Average working pressure, 180 lbs. The cost of the plant was as follows:

One Norwalk 3-stage compressor, erected	$5,180.74
Pipe-line, 4,200 ft., 5 in., including 3 charging stations	2,951.06
Two Baldwin compressed-air locomotives and fittings	4,904.33
Alterations in gangways to adapt them for locomotive haulage	665.17
Total cost	$13,701.30
Daily operating cost, for 180 days in the year	$14.69
Fixed charges, depreciation, repairs, etc., figured at 10 per cent., together with cost of steam power	9.00
Total running expenses per day	$23.69
Cost per car, at 660 cars per day	3.6 cents
Previous cost of mule haulage per car	5.1 "
Saving per year, about	$1,800.00

5. At the Wilson Colliery, of the D. & H. Coal Co., a large locomotive was installed by the Dickson Manufacturing Co., having six 26-in. drivers; wheel-base, 7 ft.; cylinders, 9 ins. × 14 ins.; gauge of track, 30 ins. The locomotive carries two tanks, 18 ft. 6 ins. and 15 ft. 6 ins. × 30 ins. diameter, with a capacity of 160 cu. ft. of air at 600 lbs. Pipe-line, 4,100 ft. long; pressure, 700 lbs. Total charging time, 1 min. 25 secs. After reduction to 125 lbs. working pressure the air is reheated. Trains usually consist of 30 cars, each weighing loaded, 5,850 lbs., though the locomotive has a capacity of 50 cars. Grades, from 9 ins. per 100 ft. against the load, to 12 ins. per 100 ft. in favor of the load. Round-trip time, for 8,200 ft. plus a switching distance of 800 ft., 16 min. Cost of haulage per ton-mile, gross, about 1¼ cents.

6. The Anaconda Copper Mine, Butte, Mont., is provided with a number of compressed-air locomotives with 5-in. × 10-in. cylinders and weighing 10,000 lbs. Over all dimensions: height, 58 ins.; width, 58 ins.; length, 10 ft. 4½ ins.; four driving wheels, 23 ins. diam.; wheel-base, 36 ins., designed for curves of 12-ft. radius. Capacity of main tank, 47 cu. ft.; pressure, 550 lbs. working pressure, 125 lbs.; charging time, 60 secs. Length of haul, 2,400 ft. round trip; load, 6 cars, weighing loaded 3,450 lbs.

each; track nearly level. The locomotives are designed to make 2 round trips, or 4,800 ft. on 1 charge, with cold air; but, by reheating with hot water, 3 round trips can be made.

At the new reduction works of the Anaconda Company, there are 13 H. K. Porter locomotives, employed in handling the products between the different divisions of the plant, which covers roughly an area of 2,200 × 2,300 ft., the length of haul ranging from 1,000 to 7,000 ft. Twelve of the locomotives have the following dimensions; weight, 26,000 lbs.; cylinders, $9\frac{1}{2} \times 14$ ins.; driving wheels, 28 ins.; wheel-base, 54 ins.; main tanks, 132 cu. ft.; draw-bar capacity, 5,700 lbs. Another locomotive weighs 42,000 lbs.; cylinders 12 × 18 ins.; driving wheels, 36 ins.; wheel-base, 60 ins.; main tanks, 218 cu. ft.; draw-bar pull, 9,180 lbs. Tank pressure, 700 to 800 lbs.; working pressure, 150 lbs.*

7. The Homestake Mining Co., Lead, S. D., employ underground 10 H. K. Porter locomotives, weighing 9,500 lbs. and measuring over all, 4 ft. 11 ins. high × 3 ft. $3\frac{1}{2}$ ins. wide × 10 ft. 6 ins. long. Gauge of track, 18 ins. They have a detachable rear end (similar to those of the Loretto Iron Co., mentioned in the 5th column of Table LVIII) to permit of transferring them from level to level, on a cage with a 9-ft. platform. At the same mine a small locomotive, with 5 × 8-in. cylinders (see Table LVIII) has been recently installed. This size is found more satisfactory, for the underground conditions prevailing in the mine, than the larger locomotive, with 6 × 10-in. cylinders.

8. Several 4-cylinder, Vauclain compound air locomotives, built by the Baldwin Locomotive Works, are in use in one of the collieries of the P. & R. C. & I. Co.† Cylinders 5 and 8 ins. × 12 ins. stroke, with valves of the balanced-piston type. Tank pressure, 600 lbs.; working pressure, 200 lbs. Driving wheels, 24 ins.; wheel-base, 48 ins.; storage tanks, of $\frac{9}{16}$ in. plate, 11 ft. $4\frac{1}{2}$ ins. and 13 ft. $7\frac{1}{2}$ ins. × 31 ins. diameter; auxiliary reservoir, 8 ins. diam. × 7 ft. 4 ins. long. Over all dimensions: 6 ft. 4 ins. wide ×

* A detailed description of this haulage plant is given by C. B. Hodges, *Cassier's Magazine*, 1905.

† *Engineering and Mining Journal*, Aug. 19th, 1899, p. 218.

14 ft. long×6 ft. 6 ins. high; weight, 22,000 lbs. Trains of 32 cars, each weighing loaded about 9,000 lbs., are hauled on 1⅔-per cent. grade, in favor of the load.

9. At the Aragon Iron Mine, Norway, Mich., is an H. K. Porter locomotive. Weight, 7 tons; height, 5 ft. 2 ins.; width, 4 ft. 2 ins.; length, 12 ft., over all. Four 24-in. drivers; wheel-base, 48 ins.; gauge, 22½ ins.; cylinders, 7×12 ins.; tank pressure, 700 lbs.; working pressure, 140 lbs.; charging time, 30 to 60 secs. Haulage distance, from 1,200 to 4,000 ft.; pipe-line, 1,800 ft.; including 750 ft. down the shaft. Locomotive hauls four 20-car trains per 10 hrs., from each of 10 loading places. Weight of loaded train, including locomotive, 43 tons; weight empty train, 18 tons. Compressed air is furnished by a Norwalk 3-stage charger: steam cylinders, 14×16 ins.; air cylinders, 10½, 7¾, and 2⅝ ins.× 16 ins., compressing 125 cu. ft. free air per minute to 800 lbs. At the compressor there are two receiver storage tanks, each 3×17 ft.

10. Compressed-air haulage plant at No. 6 Colliery of the Susquehanna Coal Co., at Glen Lyon, Penn. Following is an abstract of tests made by J. H. Bowden and R. V. Norris.* Though the plant is not of the latest pattern, the results given will be found useful.

Compressor: Norwalk, 3-stage; steam cylinder, 20×24 ins.; air cylinders, 12½, 9½, and 5 ins.×24 ins.; capacity, at 100 revolutions, 296 cu. ft. free air per minute, compressed to 600 lbs. Main pipe-line at No. 6 shaft, 4,380 ft. long, 5 ins. diameter, with 5 charging stations, and capacity of 608 cu. ft. Branch line, in No. 6 slope, 3,100 ft. long, 3 ins. diam., with 3 charging stations, and capacity of 159 cu. ft.

Locomotives: two, by H. K. Porter Co.; weight, 8 tons; tank capacity, 130 cu. ft.; pressure, 550 lbs. reduced to 160 lbs. in an 8-in. auxiliary reservoir, of 4.2 cu. ft. capacity. Cylinders, 7×14 ins.; four 24-in. drivers.

At No. 6 shaft the run averages 4,000 ft. each way, on ½ to 2¾-per cent. grades, averaging about 1 per cent. in favor of the load.

* *Transactions American Institute of Mining Engineers*, Vol. XXX, p. 566.

Run at No. 6 slope averages 2,100 ft., with nearly the same grades. Cars weigh 2,800 lbs. empty, and about 9,800 lbs. loaded, and are hauled in trains of 12 to 20. The shaft locomotive hauls about 355, and the slope locomotive 320 cars, per 10 hours, doing the work of 32 mules. Tests on the compressor showed 150 indicated horse-power at 131 revolutions, compressing 387.8 cu. ft. free air per minute.

The combined capacity of both pipe-lines is 767 cu. ft., which, at 600 lbs. gauge pressure, is equivalent to 32,500 cu. ft. free air. Capacity of locomotive main and auxiliary tanks, 134.6 cu. ft. At 508 lbs. (at which pressure the tanks equalize with the mains, the initial pressure being 600 lbs.), this is equivalent to 4,845 cu. ft. free air. In standing 12 hours, the pipe-line pressure falls to 350 lbs., the loss per hour from leakage thus being 974 cu. ft. free air, or 4.18 per cent. of total air compressed.

Table LX

Air Consumption

	Shaft Loco.		Slope Loco.
	No. 2 Plane.	No. 3 Plane.	
Number of trips, empty	3	10	16
Number of trips, loaded	3	10	15
Average number cars per trip, empty	15.33	12.7	11.4
Average number cars per trip, loaded	13	13	11.3
Average cu. ft. free air per trip, empty	1,724	5,686	1,230
Average cu. ft. free air per trip, loaded	1,631	1,898	599
Average cu. ft. free air per round trip	3,355	7,584	1,829
Average cu. ft. free air per ton-mile, on gross tonnage	113		71
Average cu. ft. free air per ton-mile, on net tonnage	203		128

Average volume free air used by both locomotives per ton-mile was: gross, 100 cu. ft.; net, 180 cu. ft. The greater quantity of air used by the shaft locomotives is due to the heavier grades and switching required. Another test showed a total consumption of 223,020 cu. ft. free air, for hauling 676 cars per day. The volume of free air apparently compressed for this work was 279,200 cu.

ft., of which 83.4 per cent. is accounted for, leaving 16.6 per cent. for leakage and slip in the compressor, leakage in air lines, and changes in temperature.

The cost of the plant, omitting boilers, was:

Compressor and extras		$2,955.75
Two locomotives and extras		5,869.76
Pipe-line: 5-in. line, 6,000 ft.	$2,914.32	
3-in. line, 4,000 ft.	1,240.46	4,154.78
Steam connections to compressor		278.27
Material and labor, compressor house and foundations, installing pipe-line, etc.		1,525.23
Charging stations		372.21
Total cost		$15,156.00

The average cost of operation of entire plant, for 2 years, on basis of 170 days' work per year, was $12.60 per 10-hour shift, including an allowance of $2.32 for steam for compressor, furnished by main boiler plant of mine. Adding proportion of fixed charges, with interest, depreciation and repairs, the daily cost, on basis of 300 days' work per year, would be $18.52 per day. For the 2 years, the average cost per ton-mile was as follows:

TABLE LXI

OPERATING COSTS

	1897 (179 DAYS).			1898 (160 DAYS).		
	Daily Ton-Miles.	Daily Cost.	Cost per Ton-Mile, Cents.	Daily Ton-Miles.	Daily Cost.	Cost per Ton-Mile, Cents.
Shaft locomotive, gross tonnage	1,485	$11.12	0.75	1,521	$12.00	0.79
Shaft locomotive, net tonnage	825	11.12	1.35	845	12.00	1.42
Slope locomotive, gross tonnage	648	11.12	1.72	720	12.00	1.67
Slope locomotive, net tonnage	360	11.12	3.09	400	12.00	3.00
Both locomotives, gross tonnage	2,133	22.23	1.05	2,241	24.01	1.07
Both locomotives, net tonnage	1,185	22.23	1.89	1,245	24.01	1.93

In these two years the saving over the expense of the mule

haulage, previously employed, was $14,218.00, or nearly the total cost of the haulage plant.

11. Following is the cost of a large colliery plant, as given by Beverly S. Randolph,* who installed and afterward operated it:

Three-stage, compound compressor		$5,300.
Pipe line: 5,600 ft., 5 in	$5,600.	
3,100 ft., 2½ in	1,700.	
1,000 ft., 1½ in	300.	
		7,600.
Two main locomotives, weight 30,000 lbs		6,000.
Five gathering-locomotives, weight 8,000 lbs		10,000.
Two boilers, each 80-horse-power		1,000.
Installation, labor, and material		4,000.
Total cost		$33,900.

This plant includes an unusually large number of small gathering-locomotives, for collecting cars from the individual workings and making them up into trains for the main haulage lines. If the locomotive equipment had consisted of four 25,000-lb. engines, costing, say, $2,800 each, and which would do the same work, the total cost of the plant would be reduced to $29,100. This cost compares very favorably with that of electric-haulage plants of the same capacity.

* *Transactions Institution of Mining Engineers* (England), Vol. XXVII (1904), p. 433.

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